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EDITORIAL NOTES

This present issue of the JOURNAL concludes the sixtieth year of continuous publication.

The JOURNAL is the vehicle by which the Society effectuates its purpose "to promote a knowledge of marine engineering and naval architecture".

The four issues of Volume 50, published in 1938, contained a four-part paper "Fifty Years of Naval Engineering in Retrospect" by Mr. H. N. Neuhaus. That paper made very evident the remarkable progress which had been made during the first fifty years of the Society in its fields of interest.

Those of us now reading have lived through the last decade of engineering progress and do not have to be told that those ten

years outshine any predecessor period in accomplishment. Even accepted fundamentals in some cases have been replaced by new concepts.

The world war which occupied about half of this decade brought the naval engineer into close contact with production engineering and its problems. In fact, he became dependent on production of the most unimaginable scope for getting his needs translated into ships in the combat areas.

Some of us have come to think narrowly of the naval engineer as being the E.D.O. This is not the case nor has there ever been any thought that either membership in the Society or interest in the JOURNAL should be limited to this specialized group. Naval Engineering extends from the design of a ship and its components to its operation and repair throughout its life. It is a field in which every line naval officer, every line Coast Guard officer, many Army and Air Corps officers, many civilians in the Navy Department and many many civilians throughout industry and business participate more or less during their careers.

The Society has always acknowledged the wide field by opening its membership to all of these groups. Any regular or reserve of the Navy, Coast Guard or Marine Corps is eligible for *Naval* membership. Any person in civil life whose knowledge of engineering is such that he can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as a *Civil* member. Any officer of the Army and any civilian who is interested in Naval matters or the merchant marine may be eligible as an *Associate* member.

While the membership of the Society has been available to large numbers of individuals, there is some question as to whether its JOURNAL has presented to prospective members enough of value to interest them and to cause their joining the Society. The JOURNAL is the only tangible asset which a member receives. The honor of belonging to a Society which has included so many brilliant and famous members over its sixty years of life palls if the current evidence of its vitality and usefulness fades.

A definite program initiated by the retiring President, Rear Admiral Roger Warde Paine, U. S. Navy, is now underway to

revitalize the JOURNAL. Thus it is hoped that the Society's prestige will be enhanced through the public evidence of its publication. The Council wishes to make membership in the Society a personal "must" to every person who can prove eligibility. It is felt that this is possible only if regular receipt and reading of the JOURNAL is a "must".

THE FORMAT

The format, or make-up, or style of the JOURNAL is of some importance in determining its value. From time to time in the past minor changes have been made but, in general, the style has retained the original conservatism. A certain amount of conservatism is proper to a JOURNAL which is not intended to make money. It is not designed for news-stand sale nor is any such popularization contemplated. However, any tricks of format which can improve the readability of the contents or the interest of the volume as a whole are desirable.

A few innovations have been included in this issue. The two principal ones which it is believed will enhance the JOURNAL'S value are the enlarged type used for the reprinted articles and the biographical sketches of authors.

REPRINTED ARTICLES

Reprinted articles are culled from most contemporary current technical publications. They can be described as papers which the editor of the JOURNAL would have liked to have had the opportunity of original publication, but some other editor had the fortunate privilege of getting them first. They are believed to be of value and interest to members of the Society and are reprinted for the convenient reading by members. The practice has been for the editorial staff to select the articles for reprinting. However, suggestions from members in this regard would be very welcome.

These reprinted articles, formerly printed in small type as "Notes", have now been elevated to the same format level as original articles. The two parts of the JOURNAL complement each other in carrying out its purpose.

Biographical Sketches

Although authors of original papers which are published receive modest payment for their effort, this is a secondary consideration for naval engineers. The most impelling reason for writing for publication is the urge to make a contribution to the profession. Such a reason deserves the recognition of publishing a short description of the author so that he can know that *his* contribution is realized and so that the reader can appraise the value of the contribution.

Photographs

Occasional photographs have been published in the Journal in addition to those used by authors for illustrating their articles. Selection of pictures of interest in a publication such as the Journal is not an easy editorial task. Submission of photographs which are considered to be of interest will be greatly appreciated.

The few photographs which are included in this issue were selected to provide to the imaginative reader a tickler for a mental review of the progress which has been made during the Society's sixty years. On the starting side of this review are the U.S.S. *Baltimore*, the horizontal engine, a Navy Yard of the 19th century. Representing today are the U.S.S. *Worcester*, the gas turbine, and the same Navy Yard in its present guise as a Naval Shipyard.

The older readers can bridge the gap between the two eras from memory and experience. The younger ones will need little imagination to fill in the glories of the professional progress from 1888 to 1948. They can then set their sights on ten, twenty and thirty years ahead and visualize the scope, if not the nature, of what their own contributions will be.

Original Articles

It is certainly apparent that no matter what effort an editor makes to improve format, he will wind up with an empty shell unless he has a selection of valuable original articles. He may be

able to do a little toward attracting authors to write and to submit the fruits of their writing, but the real incentive is the urge to present a thought to the world, to see one's own ideas in black and white and to know that others can see them too.

Every person who has been moved by the sight of the title of this magazine to open its pages is an embryo author. Most of them have had experiences or have ideas which would interest others. To the extent that they make available to the editor these experiences or these ideas will the true value of the JOURNAL reflect the ideals of the fifteen Naval engineers who started it sixty years ago.

The Council would appreciate suggestions as to how potential authors can be made to produce and also how producing authors of articles suitable for the JOURNAL can be induced to submit them to the Society rather than to some other publication.

It is realized that security regulations prevent the publication of much of the engineering and scientific details of progress since the fifty year review was published, in 1938. In spite of security there is much that can be published. It is the sincere hope of the Council that sufficient interest in the success of the JOURNAL can be evoked to bring forth papers which will describe hitherto unpublished facts which should be recorded to make the history of Naval engineering more complete.

It is also believed that engineers are thinking just as much today as in the past. Many ideas being turned over are not ready for official presentation, but have great value for bringing forth comment and discussion. The JOURNAL aspires to be the forum for such presentations and discussions.

Such formal rules as have been established will be found under Association Notes in the after end of this magazine.

THE PRESIDENT

For the second time in its history, the Council of the Society had the experience of accepting the resignation of its President. Entirely beyond his or the Society's control, Admiral Paine has been transferred from Washington to California. His removal

from the Washington area makes him ineligible to continue in the office to which we elected him.

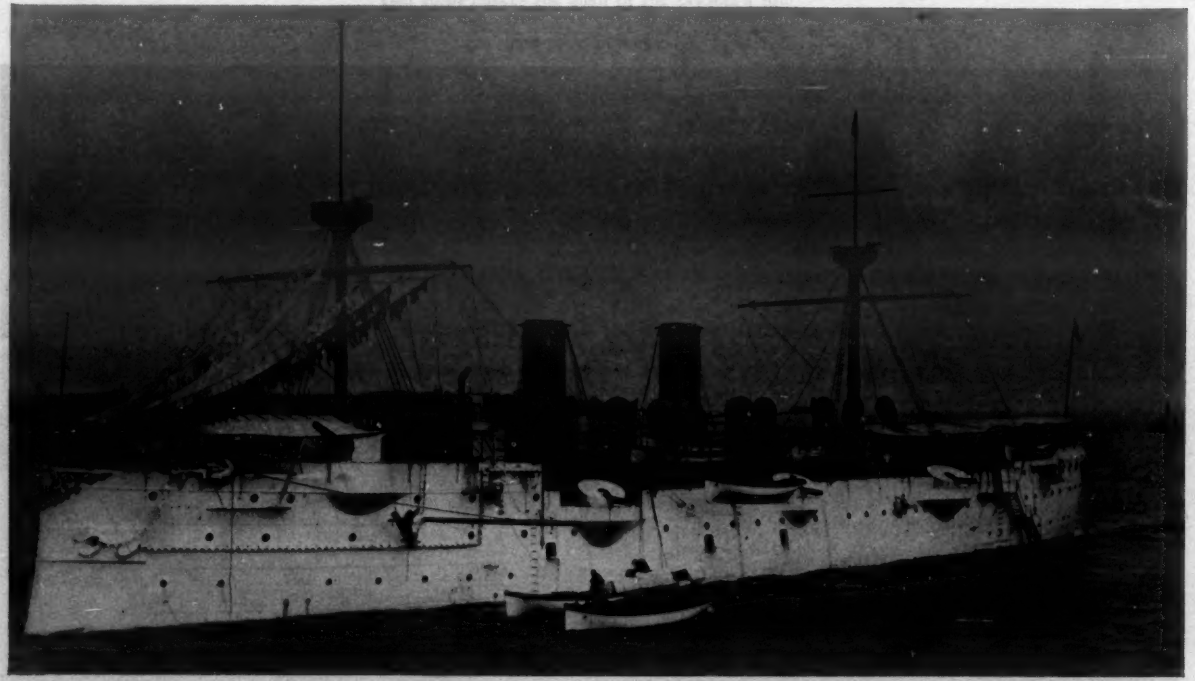
Admiral Paine need not be presented to the members. They know him. His service as Secretary-Treasurer and as a member of the Council prior to his election to the Presidency, coupled with his non-Society duties and activities have marked him as one of the leaders of the Society and of the Naval Engineering profession. The Council can report to the membership that Admiral Paine has been a dynamic President who has thoroughly demonstrated the Society's wisdom in conferring the office upon him.

It is regretted that Admiral Paine was unable to complete his term. It is certain, however, that his transfer will not remove his interest in the Society nor discontinue his efforts to improve it.

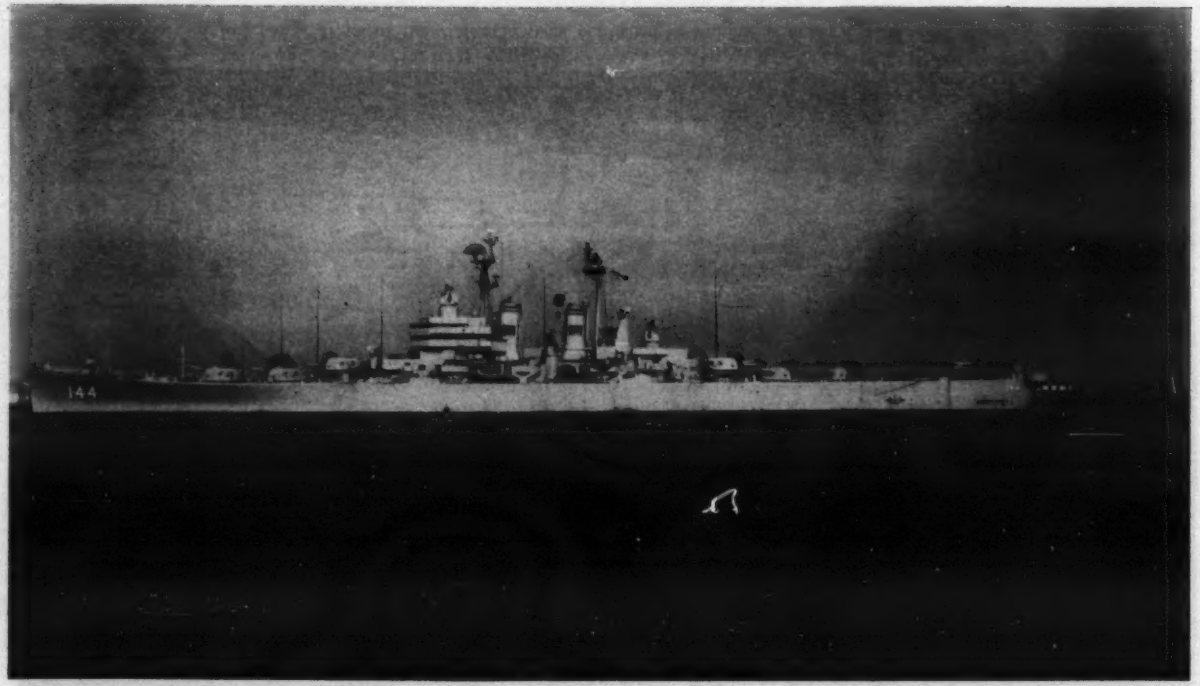
Read Admiral Harvey F. Johnson, U.S.C.G. (Retired), a past-President, was appointed President for the remainder of the year.

Rear Admiral Claud A. Jones, U.S.N.

It is with sincere regret that the Society announces the death of a Past President and an outstanding member of the Naval Engineering profession, Rear Admiral Claud A. Jones, U.S.N.



U. S. S. *Baltimore.* CRUISER NUMBER 3.
The report of full power trials of the *Baltimore* was published in Volume Number 1 of the JOURNAL.



U.S.S. *Worcester*

This latest ship to join the U. S. Fleet may be compared with the photograph of the *Baltimore*. The period between these two ships covers the span of the Society's life.

This latest ship to join the U. S. Fleet may be compared with the photograph of the *Baltimore*. The period between these two ships covers the span of the Society's life.

PART I.

(Original Articles)

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part I are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

A NEW CONSIDERATION FOR WARSHIP DESIGN.

By Commander Edw. J. Fahy, U.S.N.

The evolution of modern warships from the days of the first U.S.S. *Monitor* to and through the last world war, has been chiefly determined by the factors of armor, armament, propulsion, and damage control. In essence the warship must be able to strike offensively, to endure defensively, and to be at the proper place at the proper time.

Such capabilities, however, leave out certain considerations which, in modern warfare, are equally as important as offensive power, defensive power, mobility, and invulnerability. These considerations are covered by the term Naval Electronics.

Under Naval Electronics are found radio, radar, and sonar. Each of these includes many subdivisions such as offensive and passive countermeasures, identification, homing and guidance systems, infrared systems, target range and bearing determination, etc., through the entire gamut of the military services of communication, detection, and ranging.

By the very nature of the science of electronics, difficulties are at once presented to the warship designer. Boilers and turbines can be located judiciously and, in the case of electric drive ships,

The Author.

Commander Edward Joseph Fahy, U.S.N., graduated from the United States Naval Academy in 1934.

After three years in heavy cruisers, he entered submarine service in 1937. He took two year postgraduate course in Electronics Engineering.

He commanded U.S.S. *Plunger* in the Pacific during World War II. During the last year of the war served on the staff of Commander Submarine Training Command, Pacific as New Developments — Electronics Officer.

Presently in charge of the Ship Installation and Maintenance Branch of the Electronics Division of BuShips. Duties involve not only responsibility for installation and maintenance of shipboard electronics equipment for U. S. Fleet, but also the design planning for electronics installations incident to new construction, conversions, and preliminary ship designs.

Awards—Legion of Merit, two letters of commendation with ribbons, with combat "V"s, and Navy Unit Commendation for war patrols in U.S.S. *Plunger*.

with great flexibility. Master gyro compasses, and various motor generator power supplies can be ingeniously fitted into various available spaces. Pumps can be installed in convenient areas. Guns can be scattered throughout the topside with almost reckless abandon in order to provide a continuous hemisphere of anti-aircraft fire. Armor can be placed overall to effect a sturdy, boxlike rigidity.

But naval electronics equipments must operate in conjunction with external antennas whose function is to radiate energy into space, or to receive energy from space. (A sonar projector or transducer may be considered an antenna, i.e. a radiator and collector of energy, operating in sea water instead of free space.)

The fact that antennas are necessary for the operation of electronics equipment places the shipboard requirements for installation of this equipment in a very unique category. For, as a result, the locations within the ship of the transmitting and receiving equipment components are governed not only by space, weight, and usage factors, but also by the necessity for delivering maximum power to, or absorbing maximum signal strength from the equipment antenna.

It is the antenna and the critical conditions which affect it that have given rise to a new consideration in warship design.

Since the antenna is the consideration that differentiates electronic equipment from electrical and mechanical shipboard equipment, it might be well, at this point, to examine the bearing and importance of the antenna on the overall efficiency and performance of an electronic system.

If a normally functioning radio transmitter is operated in a location ashore, free from structural (and hence, reradiating) interference, with an antenna and transmission line properly designed for the power range and for a specific frequency, the expected radiation efficiency will be very high, and in the case of a high-frequency antenna will be greater than 90 per cent.

On the other hand, measurements made of shipboard antennas have shown that a transmitter using some presently installed shipboard antennas can have an antenna system radiation efficiency of only 0.2 per cent! This problem is particularly harassing if low frequency operation is necessary such as is required in arctic regions. Our present shipboard installations, which are not rigged with horizontal "flattop" antennas but must

depend on vertical wire antennas (even if over 90 feet in length) can expect radiation efficiencies of 0.02 per cent at 175 kc to 0.25 per cent at 550 kc!

With efficiencies like these it is evident that mere "brute force" methods will not assure the range of communication desired. Installation of a two thousand watt transmitter in place of an existing 500 watt equipment would result in such a slight gain in radiated power that the additional space, weight, and power required for the much larger transmitter could not in any way be justified.

If however, the same transmitter is fed into a horizontal shipboard flattop antenna (about 95 feet long, with a 69 foot leadin, for example) the radiation efficiencies will be increased to the values 0.25 per cent to 12.5 per cent over the same range of frequencies listed above. The addition of a sloping after section of 150 feet added to the horizontal flattop (i.e. run from the mainmast yardarm supporting the flattop, to a spot back aft on topside) will result in a further gain of radiation efficiencies to 0.8 per cent and 33.7 per cent respectively!

To state the foregoing in a different manner, let P be the power which is supplied to the antenna by the 500 watt transmitter. Assume that the amount of power supplied to the antenna by the 2000 watt "brute force" transmitter is 4P. The power radiated would, of course, be increased four times. But considering the efficiencies involved when using the 90 foot vertical wire, the power radiated has only increased from .0002P to .0008P at 175 kc and from .0025P to .01P at 550 kc. When the flattop antenna is used with the 500 watt transmitter the power radiated increases to .0025P at 175 kc and .125P at 550 kc or 3 to 12.5 times more than was obtained by using the 2000 watt transmitter with the vertical wire antenna. If the 150 foot after section is added, the radiated power obtained from the 500 watt equipment is 10 to 33.7 times greater than that obtained from the 2000 watt "brute force" transmitter!

In most cases the shipboard antenna will produce an undesirable directionality pattern (because of proximity to most structures and other antennas) which further reduces its usefulness as an omnidirectional radiator, and also lessens, by a distinctly measurable amount, the available radiation energy. Naval communications must be carried on at long range, with an extremely

high degree of reliability, under varying atmospheric and climactic conditions, and in most cases, with omnidirectional coverage.

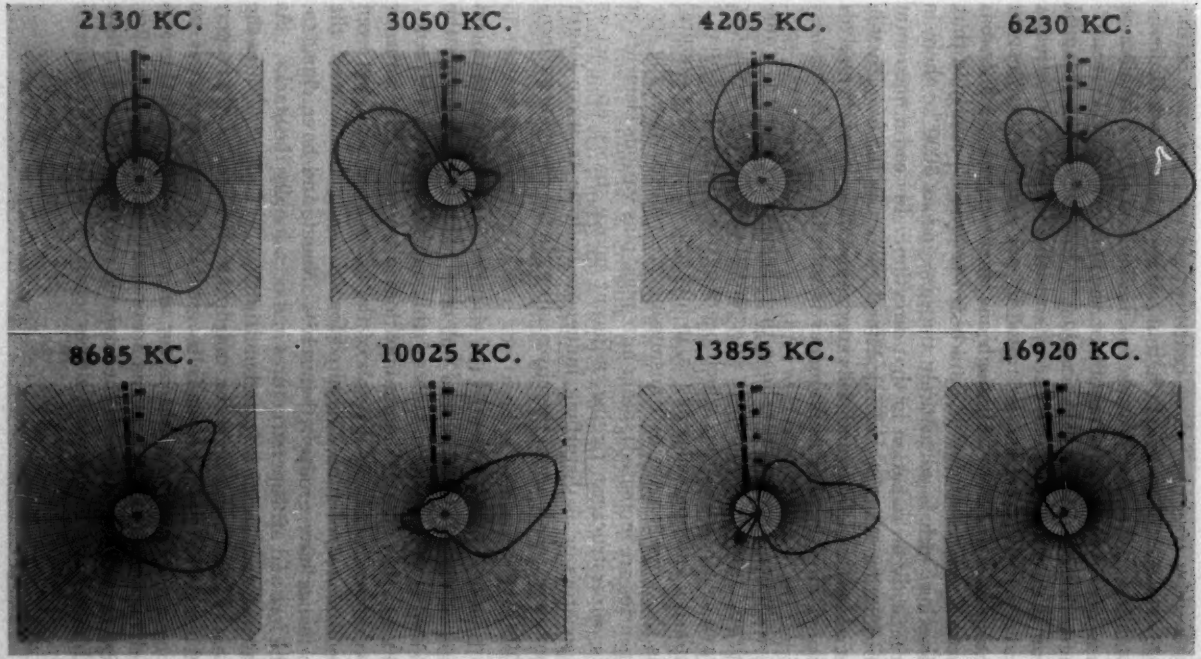
Consequently, in transmitting, an adequately powerful signal must be radiated into the ether. When receiving, the signal which exists in the vicinity of the antenna must be efficiently absorbed and presented to the receiver for detection and translation into useful intelligence. In naval communications these services must be supplied over an extremely varied number of frequencies which cover almost the entire electromagnetic spectrum. This requirement further complicates the problem of supplying an efficient radiator, because as the frequency of communication is varied, the physical dimensions of the antenna should vary in order to provide the most efficient radiator of energy.

Now if the antenna which is under consideration could be made to operate efficiently over a wide range of frequencies, the problem in itself would be extremely complex and difficult, even if it were in a location ashore free from all structural and other reradiating interferences. But when the antenna is installed in a warship, so that it must be close to a very heterogeneous superstructure, must be built to fit the usual warship limitations of compactness, ruggedness, ability to withstand the working of the ship and the shock of gunfire, be clear of arcs of gunfire, etc., the problem for the warship designer and the electronics engineer is indeed complicated. If, moreover, not one but several dozen of these antennas must be provided, as is the case in large combatant ships, the problem appears almost hopeless.

An excellent example of what this situation is actually like may be seen from the following real case:

A certain heavy cruiser class ship had 67 antennas for various electronic functions such as communications, countermeasures, radar, etc. One of the communications antennas was located at frame 66 mounted three feet outboard of the starboard flag bridge, and was used as a transmitting antenna. The result desired, of course, is maximum omnidirectional transmission. The actual result, at various frequencies of transmission, is shown in the series of patterns obtained by the U. S. Navy Electronics Laboratory at San Diego, California.

Were omnidirectionality achieved, the patterns obtained would



DIRECTIVITY PATTERNS OF HEAVY CRUISER MF-HF TRANSMITTING ANTENNA (35′ WHIP). ALL PATTERNS WERE TAKEN AT ZERO DEGREES ELEVATION.

be a series of circles whose center would be the center of the polar chart. However, the actual patterns show marked directionality characteristics whose major direction shifts with change in frequency. As a matter of fact, the pattern at 2000 kc, resulting in a fore and aft directionality, is similar to that obtained from a directional antenna array consisting of a driven antenna and a parasitic antenna, separated by one quarter wave length, acting as a director. Although one might expect the antenna, located well forward and to starboard to "shoot" clearly in that direction, such is not entirely the case and at 3000 kc there is practically no radiation in that direction. Instead a maximum of radiation exists on bearing 320 relative! As the frequency is increased, large lobes and nulls appear and these lobes themselves change directionality as frequency is varied.

A study of these results shows that the distance from the transmitting antenna to the after superstructure is roughly a quarter wave length at 2000 kc, which bears out the observation that a parasitic antenna exists and is, in fact, the after superstructure! As the wave lengths of transmission are reduced (i.e., the frequencies are increased), other antennas, plus various elements of the superstructure, rigging, adjacent smoke stacks, etc., act as resonators or as disturbing influences both as to wave length and phase relationship, thereby resulting in the changing patterns shown.

Imagine how complicated the situation is for the communication officer of the ship who not only has to worry about the variation of lobes and nulls of this one antenna as frequency changes but has 51 communication antennas in the same or worse condition! As a result of the widespread arcs, and even hemispheres, of gunfire required for defense against aircraft, the antennas have grouped themselves around, and have clung to, the fore and aft superstructures and occasionally the stacks. As the antennas shrink into and multiply on the superstructures the point is reached where the magnitude of effect of the superstructure on the antenna is so great that the antenna is no longer a free radiator but is, for all practical considerations, a driving element in an antenna array of which the superstructure, rigging, stacks, etc., are the major elements wihch contribute unknown and unmeasured effects.

The discussion to this point has been concerned with communication antennas, and with other omnidirectional types such as radio and radar beacons, IFF, countermeasures intercept, etc.

In the radar field, the considerations differ somewhat because of the usual directionality of radar antennas. Since search radar antennas are operated essentially like searchlights and demand 360 degree coverage, the problem becomes complicated when several radar antennas must be installed, all requiring installation at the highest point of the ship in order to obtain unrestricted coverage. Since warship construction progresses only on a series of compromises, inevitably one radar antenna achieves the desired spot and the others, in compromise positions, try to "look through" masts and other parts of the superstructure with resultant blind spots in the coverage desired.

At one time during the war there were seven different electronic equipments which had as an installation requirement that the antenna be mounted at the highest point of the ship, entirely in the clear. Obviously only some form of overgrown merry-go-round or a return to the seven-masted schooner could solve this problem. Moreover, in addition to the foregoing considerations the radar antenna is growing to such a size and weight that masts and even tripods are no longer adequate to carry it. There are presently, on aircraft carriers, radars whose antenna alone weighs just under three and a half tons and requires a swing circle of 19 feet in diameter! And the trend is to much bigger, and hence better, radar antennas for all warships, including the destroyer with its single superstructure pile and foremast.

Further, considering the underwater sound (sonar) projector or transducer, it is seen again that Naval electronics equipment must become integral with the warship. The problem was virtually nonexistent when the only piece of sound apparatus on major ships was the sonic fathometer which required a single, skin mounted projector, not on the center line. Today the growth of the field of underwater detection and attack necessitates the installation of several shaft mounted projectors or transducers which must be lowered well clear of the ship's hull and be so located as to give optimum performance.

Unfortunately for underwater sonar performance on warships, the equipment must always work against its own better performance. The best sonar location would be in a noiseless craft,

with the transducer immersed well below the depth at which water noise from the surface of the ocean would be a factor, and which was used when this noiseless craft were making no speed through the water.

Even if sailing craft or possibly Flettner rotor ships were used in an effort to obtain the noiseless craft, the ever present requirement that the sonar equipment must be carried into direct and launch attacks against fast moving underwater targets results in high, noise-creating speeds through the water. Since the amount of noise increases with the increase in speed, the result is not only a reduction in sonar performance, but if the speed is increased beyond a certain point the sonar equipment will be rendered ineffective.

One foreign navy has rigged a sonar projector on a boom which extends underwater forward from the ship's forefoot in an attempt to obtain freedom from ship noise, cavitation, etc. While theoretically better performance is obtained, this rig has the very serious objection of being a fair weather job because the effects of pitching in a rough sea would be maximum in this position. The same objections as to location are found in destroyer escort (DE) hulls which had the sonar projector well forward. The pitching of the ship in a rough sea rendered the projectors almost useless.

The German Navy was able to achieve very good results from a sonar installation by fairing a full arc of hydrophones into the hull itself in a very well chosen location, with the entire area of the hull in the vicinity of the hydrophones carefully faired and "clean".

When an array of several sonar projectors in one hull is called for, and all of these projectors are to be used at the same time, and all are seeking the optimum position on the hull, the problem is no longer simple, nor can past experience be used except very generally. Although the antenna (and sonar transducer) problem is extremely complex, it is not without hope of solution.

It is true that on existing warships the improvements which can be effected are slight unless the ship is to be redesigned. The uses and demands for electronics made such gigantic strides during the last war that conventional ship type designs lost their clean superstructures to a heterogeneous jumble of whip antennas spike antennas, radar "dishes", radar "mattresses", racons,

homing beacons, fire control radar "polyrods", and endless dipoles, "Y" antennas, crossed loops and just plain wire antennas. As new developments arose they were simply added to the existing structure.

It is in new ship designs or in extensive ship type conversions that the greatest opportunity for improvement lies. This improvement can be effected by designing the superstructure to fit the antenna services with which the ship must be provided. Since it is known that a mast 120 feet in height resonates at about two megacycles, it may be possible, especially at the lower frequencies to design the masts themselves for use as antennas! It may also be possible to design various other parts of the ship's superstructure for use as antennas in addition to fulfilling their existing functions.

University of California engineers working on antenna studies under a navy contract, found that exciting the mast of their PC test boat resulted in a better antenna pattern than that obtained from the regularly installed whip antenna!

Actual antenna pattern measurements of a 692 class destroyer taken by the U. S. Navy Electronics Laboratory, San Diego, showed that the pattern of a whip antenna, located, perforce, in close proximity to a 40 millimeter gun mount, changed radically as the gun mount was trained and elevated. Since the interrelationship between antennas and structures is so complex, resort must be had to empirical means to obtain a solution. The model study techniques employed at the David Taylor Model Basin and at the Ship Antenna Systems Model Range at the U. S. Navy Electronics Laboratory, San Diego, are ideally suited to this problem. The David Taylor Model Basin should prove invaluable in the solution of the problems incident to location of sonar transducers on a ship's hull, and to the shaping of the hull itself.

Noise, both water noise and background noise, is one of the chief factors in limiting sonar performance. Noise is readily created by discontinuities in the hull shape as it is driven through the water. Hence, for better sonar performance, "sonar hulls" must be designed to achieve that end. The undersea warfare sonar problem is so intricate, and the demands made are so exacting and beyond present actualities that the time is at hand when the ship must be designed around the sonar equipment. As more is learned about undersea hydrography and as further

surveys of oceans are completed, sonar equipment and ships should be developed to utilize the results of those surveys. It is no longer feasible to build the ship and then hang on the sonar equipment like so many ancillary appendages. The optimum location of the sonar equipment should be determined first and the interior of the ship planned around it.

The U. S. Navy Electronics Laboratory set up a ship antenna systems model range in August 1944 to study the causes and possible reduction of ship communications antenna directivity. Ship models, constructed of sheet brass, are built with precise detail of structure, with rigging carefully matched to the prototype, on a scale of one to twenty-four or one to forty-eight. The model "ocean" consists of half-inch galvanized mesh spread over a flat asphalt pavement, in radial overlapping strips, bonded together, and covering a circular area approximately 160 feet in diameter. The ship model is centrally located in this ocean and directivity patterns are taken by rotating the model at constant angular velocity. Wave length, resistivity and model dimensions are scaled using the same factor. Synchros are used to link a polar recorder turntable with the rotation of the model. A directive antenna, fed by a radio frequency oscillator, is located at the edge of the ocean and beams r-f- energy at the receiving antennas located on the ship model. Tests to check the validity of the model can be readily made by a comparison of antenna patterns from the model and the actual ship. Remarkable similitude is obtained, so precise is the model structure and rigging.

Here, then, is the means to effect the amalgamation of antennas and structures. As the designer works out his preliminary arrangements, a rapid, inexpensive check of both arrangements and antenna patterns can be readily made. If the design study and the model range evaluation are made simultaneously, the warship can evolve in a truly balanced manner with the services of gunnery, communications, propulsion and damage control operating as a conglomerate entity.

The consideration of antennas during the design of ship structures could have an extremely beneficial effect on not only the warship but its equipment and materials of construction. Since the multiplicity of communication equipment demands the use of a common antenna for several equipments, the development of a

good multiplex system is required. When a good multiplexing system for both transmitters and receivers is designed, the number of antennas can be reduced, and the few good antenna locations will be used to the best advantage. However, for multiplexing, a broad band antenna is a must. This might take the form of multiwire antennas (which are less sharply tuned than single wire antennas), or some form of cylindrical radiator. It is a pity that the old battleship cage masts are no longer extant. They probably would have made excellent broadband antennas!

The warship designer, working under this new consideration, will cooperate very closely with the electronic equipment designers of radar antennas. Since the radar antennas must have an unobstructed "look" through 360 degrees, and since there are several radars per ship, a rotating (merry-go-round) antenna system could be developed, having the antennas stacked above each other, each revolving on its own track around a common central rube or mast, with an anti-aircraft gun nest surmounting the top.

Because shipboard radar antenna systems are growing to a weight of a score of tons, the most intimate cooperation must exist between the designers of the ship and the equipment, else the equipment will never be properly installed.

Again, in the design of aircraft carriers, particularly the true flattop type, the ship designer and electronics engineer may be able to utilize the flight deck itself as an antenna. By using the slot antenna principle, radiation from a flat surface can be achieved as was done in the U. S. Navy designed double-slot antennas. Portions of the flight deck might be isolated (electrically uncoupled) by high impedance, cavity-backed slots (filled with a suitable dielectric material to make a continous surface for airplane operations) and be fed electrically through the deck supports. This would have an additional advantage of being a steerable system. For "it should be of interest to notice that the double-slot antennas possess a natural characteristic which is of advantage to lobe switching. If only one of the two slots is fed, the radiation pattern will lean toward the fed slot." (Lindenblad: Slot Antennas. Proc. of I.R.E. December, 1947.)

From all of the foregoing it can be seen that the antennas for electronic systems must be as carefully integrated with the warship as any other service such as guns, propelling machinery, etc.

It is seen that close coordination between the hull designer on the one hand, and the electronics engineer on the other, can only result in the best solution for well balanced warship design.

As Fleet Admiral Nimitz has so succinctly stated: "The Navy is still the first line the enemy must hurdle either in the air or on the sea in approaching our coasts across any ocean. The earliest warning of enemy air attack against our vital centers should be provided by naval air, surface, and submarine radar pickets deployed in the vast ocean spaces which surround the continent."

Only by the closest integration of the electronic system and its antennas with the ship itself can the warship achieve that all around strength that must include reliable communications as well as fire power and mobility.

Only in this manner can there be assured the strong, alert Navy which is needed for these troublous times.

TITANIUM—A MODERN METAL.

By Julius J. Harwood, Metallurgist.
Office of Naval Research.

Introduction.

Contemporary metallurgists have the difficult task of developing metals and alloys capable of withstanding extreme temperature conditions at both ends of the temperature scale, under increasingly severe operating stresses. The problems related to the use of metals for military and civilian applications have been augmented many-fold as a result of the accelerated technological and engineering advances which have been accomplished in recent years. The realization that the ultimate solutions to many of these problems do not lie within the range of mere improvement of present day metal systems, by one means or another, has resulted in an intensified search for "special-purpose" metals among elements which were formerly categorized as "rare" or which had not been adequately exploited.

One such group of interesting metals, is the "Refractory Metals", so named because of their extremely high melting points. This group, which has lately achieved metallurgical prominence for gas turbine, rocket and guided missile materials,

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A previous article, "Metallurgical Tracers and Naval Research", describing the use of radioactive isotopes in the ONR metallurgical research program was published in the Journal of the Society of Naval Engineers in February 1948.

includes zirconium, titanium, columbium, molybdenum, tantalum and tungsten with a range of melting points from about 3100° F. to 6100° F. (1700° C. to 3370° C.). Until recent times tungsten, tantalum and molybdenum were considered the principal refractory metals and their development was closely associated with the incandescent lamp, x-ray tube and electron tube industries. There is, today, considerable interest in and research effort underway for the development of titanium and zirconium and their alloys as engineering construction materials. Because of its promising potentialities for application as both a structural metal and heat-resistant material, titanium has been singled out for discussion at this time.

The desirable characteristics of a metal or alloy which would be important for its general introduction into the class of useful structural materials are availability, producibility, light weight, high strength, corrosion and thermal resistance. These, of course, are supplemented by the specific requirements incident to a particular application. Availability and producibility include not only the existence of satisfactory amounts of raw deposits, but also the capability of being extracted and isolated from its ores and processed satisfactorily into a useful form. In the light of current strategic metals considerations, this characteristic is achieving a preeminent position in dictating the choice of materials. Although a given metal may satisfactorily fulfill a set of specified requirements, compromises must often be made in the selection of the actual material to insure a continuous supply necessary for high production needs.

What are the characteristics of titanium that warrant the extensive interest currently being directed toward its exploitation, development and research? The ores of titanium are plentiful. The metal has a density of 4.5, midway between that of iron and aluminum and even in this early developmental stage it has a strength-weight ratio approaching that of our high-strength steels. Its corrosion resistance is comparable to that of stainless steel, and its high melting point directs attention to the study of its high temperature properties. Certain of its other properties lend themselves to many special purpose applications. It may be stated that of all the metal systems that have yet been commercially exploited, titanium and its alloys bid fair to fulfill, most nearly, the requirements for an "all-purpose" metal. Little

is yet known about the unexplored field of titanium alloys, but it seems almost certain that as research and development progress, present potentialities will be transformed into actual realities.

The following discussion will describe in more detail the occurrence, manufacture and properties of titanium and its alloys and the research programs currently underway in the Navy and other government agencies aimed at furthering the understanding and development of this interesting metal.

Availability of Titanium.

Titanium is the ninth most abundant element and has been estimated to comprise about 0.65% of the earth's crust. Of more practical significance is the fact that it is the forth most abundant structural metal, being exceeded only by aluminum, iron and magnesium (Figure 1). The two most important sources of

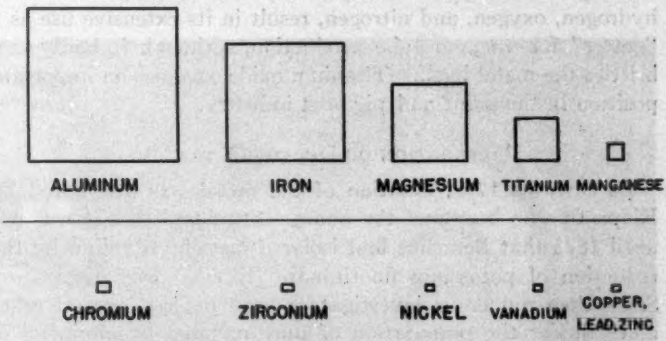


FIGURE 1.

titanium are rutile (titanium dioxide) and ilmenite, a ferrotitanate ($FeTiO_3$). Ilmenite is quite plentiful domestically, with large reserves present in the Adirondack region of New York, in Wyoming, Virginia, and Arkansas and in the beach sands of Florida East Coast. The beach sands of Travencore, India, are also rich in ilmenite and prior to World War II provided the major source of the ore for import by the United States. Since

1942, however, the United States has become the world's leading producer of ilmenite. Rutile, while not as plentiful as ilmenite, is present in substantial deposits in Virginia. There are also great quantities of titaniferous magnetite sands in Japan and titanium ore deposits in the U. S. S. R. Extensive deposits of ferro-titanium have recently been discovered in Canada. Thus, it is obvious that the "rare" classification of titanium is not due to the scarcity of its ores but primarily to the difficulties and cost of producing it in a pure state. The results of early workers indicated that the titanium metal was extremely brittle, which discouraged its exploitation as a structural metal. Titanium is, nevertheless, not a stranger in the field of metallurgy, having a wide application as an alloying element in steels and in the production of titanium carbide, a major constitutent of cutting tools. The very properties which are so deleterious to the production of pure ductile titanium make it useful to the electronics industry. The high reactivity and absorption by titanium of gases, such as hydrogen, oxygen, and nitrogen, result in its extensive use as a "getter" for vacuum tube production, although it badly embrittles the metal itself. Titanium oxide occupies an important position in the paint and pigment industry.

Preparation of Ductile Titanium.

As early as 1797, isolation of the metal was attempted by Klaproth who bestowed the name, "titanium," but it was not until 1825 that Berzelius first isolated metallic titanium by the reduction of potassium fluotitanate (K_2TiF_6) with potassium. Since then numerous investigators have devised several other methods for the preparation of pure metallic titanium, all of which fall within the following general classifications: (1) the reduction of titanium tetrachloride with an active metallic element; (2) the thermal dissociation of titanium halides; (3) the reduction of titanium oxide with reducing agents or metallic elements; (4) the electrodeposition of titanium from fused salts. The first and second methods have been most successful and are at present under commercial development.

Metallic titanium was prepared by Hunter who reduced titanium tetrachloride with sodium in an iron bomb to form practically pure titanium (99.7 to 99.9%). This process was later modified by Kroll, who substituted molten magnesium for

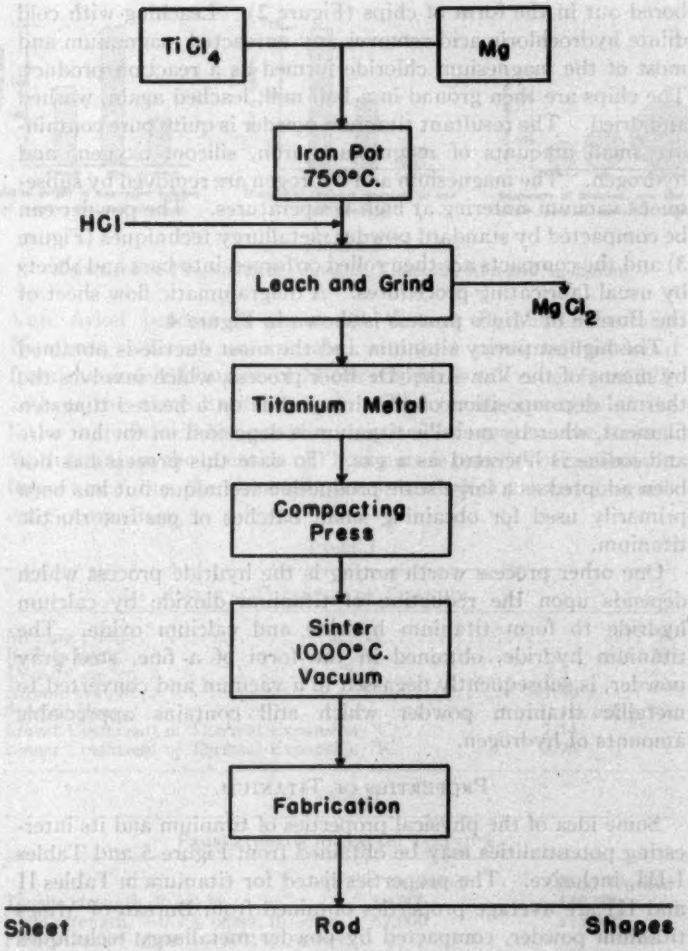
sodium, and thereby simplified the operating procedure by eliminating the necessity for the high pressures involved with the Hunter process. The Bureau of Mines, Department of the Interior, adapted the Kroll process for large-scale operations, and at the present time is operating a pilot plant at Boulder City, Nevada, capable of producing 100 lb. of titanium powder per day. Consideration is being given to the expansion of the plant capacity to permit the production of a ton of powder per day.

The Bureau of Mines' reduction process is carried out in an iron pot 12 in. in diameter and 14 in. high. After reducing any oxides which may be present on the interior surfaces, the pot is charged with about 20 lb. of magnesium. Melting of the magnesium occurs at about 750° C. under a helium atmosphere. The liquid titanium tetrachloride is introduced and allowed to drip on the magnesium. A strongly exothermic reaction occurs which maintains the temperature of the chamber at the desired point without external heating for about two-thirds of the run. After

FIGURE 2.
Solidified charge, consisting primarily of Titanium with magnesium chloride and unreacted magnesium, preparatory to boring on a lathe.
U. S. Bureau of Mines photo.

FIGURE 3.
Titanium ingot compacted by powder metallurgy technique.
U. S. Bureau of Mines photo.

the greater part of the titanium tetrachloride has reacted with the magnesium to form titanium and magnesium chloride, the temperature is raised to about 900° C. and the reaction goes to completion. The container is then cooled and the charge is



Flow Sheet of Titanium Metal Production

FIGURE 4.

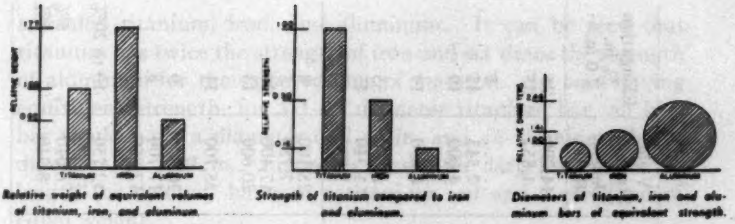
bored out in the form of chips (Figure 2). Leaching with cold dilute hydrochloric acid removes any unreacted magnesium and most of the magnesium chloride formed as a reaction product. The chips are then ground in a ball mill, leached again, washed and dried. The resultant titanium powder is quite pure containing small amounts of magnesium, iron, silicon, oxygen, and hydrogen. The magnesium and hydrogen are removed by subsequent vacuum sintering at high temperatures. The powder can be compacted by standard powder metallurgy techniques (Figure 3) and the compacts are then rolled or forged into bars and sheets by usual fabricating procedures. A diagrammatic flow sheet of the Bureau of Mines process is shown in Figure 4.

The highest purity titanium and the most ductile is obtained by means of the Van Arkel-De Boer process, which involves the thermal decomposition of titanium iodide on a heated tungsten filament, whereby metallic titanium is deposited on the hot wire and iodine is liberated as a gas. To date this process has not been adopted as a large scale production technique but has been primarily used for obtaining small batches of gas-free ductile titanium.

One other process worth noting is the hydride process which depends upon the reduction of titanium dioxide by calcium hydride to form titanium hydride and calcium oxide. The titanium hydride, obtained in the form of a fine, steel-gray powder, is subsequently degassed in a vacuum and converted to metallic titanium powder which still contains appreciable amounts of hydrogen.

Properties of Titanium.

Some idea of the physical properties of titanium and its interesting potentialities may be obtained from Figure 5 and Tables I-III, inclusive. The properties listed for titanium in Tables II and III are average properties obtained from Bureau of Mines titanium powder, compacted by powder metallurgy techniques and rolled into thin sheets. It should be remembered that titanium made by this method although relatively pure does contain small amounts of gaseous and other impurities which result in higher strength properties and lower ductility properties than pure ductile titanium, such as made by the Van Arkel process. For example, the hardness of a titanium rod produced by the



Relative weight of equivalent volumes of titanium, iron, and aluminum.

Strength of titanium compared to iron and aluminum.

Diameters of titanium, iron and aluminum bars of equivalent strength.

FIGURE 5.
Comparative Properties of Annealed Titanium, iron and aluminum.

Van Arkel process is about 90 Vickers Pyramid Hardness Numbers, whereas an annealed titanium specimen produced from the sintered Bureau of Mines compacts will range from 225-250 on the same hardness scale. The tensile strength of the latter material will be approximately twice as great as the former. With titanium the question of "how pure is pure" is of paramount importance, but the values shown nevertheless present a significant comparison.

TABLE I

PHYSICAL POPERTIES OF TITANIUM.

Melting Point	3272 °F.
Boiling Point	9212 °F.
Density gm/cc	4.5
Atomic Weight	47.90
Specific Heat	0.142 cal/g/°C.
Linear Coefficient of Thermal Expansion/°C	7.14
Linear Coefficient of Thermal Expansion/°F	3.96

TABLE II

PROPERTIES OF METALLIC TITANIUM.

	Annealed	Cold Worked
Tensile strength, lb/sq in.	80,000	120,000
Yield strength, −0.2% offset, lb/sq in.	67,000	113,000
Proportional Limit −0.01% offset, lb/sq in.	59,000	84,000
Elongation −% in 2.0 in.	25	7.5
Young's Modulus of Elasticity, lb/sq in.	16.8×10^6	15.4×10^6
Hardness-Rockwell	88B	25C
Electrical Resistivity ohm/cm³	50	52
Density, gm/cc	4.5	4.5
Crystal Structure up to 880°±20° C.	Hexagonal close packed a=2.95 Å c=4.73 Å	
Above 880° C.	Body-centered cubic a=2.32 Å	

TABLE III.
COMPARISON OF STRENGTH-WEIGHT DATA.

	Condition	Tensile Strength lb/sq in.	Yield Strength lb/sq in.*	Density	Tensile Strength / Density Ratio	Yield Strength / Density Ratio	Elongation % in. 2.0 in.
Titanium	Annealed	80,000	64,000	4.5	17,800	14,200	25
Aluminum-2S (Pure Aluminum)	Annealed	13,000	5,000	2.7	4,815	1,852	35
Magnesium	Annealed	25,000	.....	1.74	14,390	.....	5
Ingot Iron	Annealed	40,000	20,000	7.86	5,090	2,545	45
Aluminum Alloy-24SO	Annealed	27,000	11,000	2.77	9,810	4,000	19
Magnesium Alloy-M	Annealed	32,000	16,000	1.8	17,700	8,890	19
Stainless Steel	Annealed	95,000	45,000	7.91	12,000	5,690	55
Copper	Annealed	30,000	12,000	8.94	3,360	1,340	45
Brass (70Cu-30Zn)	Annealed	45,000	15,000	8.40	5,350	1,783	65
Titanium	Cold-Worked	120,000	113,000	4.5	25,700	25,100	8
Aluminum 2S	Hard-Rolled	24,000	21,000	2.7	8,890	7,780	5
24ST-Al Alloy	Heat-Treated	68,000	48,000	2.77	24,500	17,400	20
24ST-36 Al Alloy	Heat-Treated and Cold-Rolled	73,000	57,000	2.77	26,400	20,600	13
75ST-6 Al Alloy	Heat-Treated and Cold-Rolled	82,000	72,000	2.8	29,300	25,700	11
Mg Alloy AM59S	Wrought	51,000	38,000	1.82	28,000	20,800	9
Stainless Steel	Wrought	185,000	140,000	7.91	23,400	17,700	8
Ingot Iron	Cold-Worked (Max.)	100,000	100,000	7.79	12,850	12,850	..
Range of Commercial Steels	Quenched and Tempered	.....	100,000	7.8	.....	12,750	10 to 20%
Copper	Hard-Rolled	60,000	250,000 54,000	8.94	6,720	31,800 6,040	5%
Brass (70Cu-30Zn)	Hard-Rolled	105,000	75,000	8.4	12,500	8,940	5%

*0.2% Offset.

Table II indicates the good workability of titanium and the excellent properties of the cold-worked metal. The high yield strength-tensile strength ratio when combined with the low density reveals its potential usefulness as a structural metal. Figure 5 graphically illustrates the comparative properties of

annealed titanium, iron, and aluminum. It can be seen that titanium has twice the strength of iron and six times the strength of aluminum for the same volume of material. In bars having equivalent strength, for a 1-in. diameter titanium bar, an iron bar would have a diameter of 1.41 in. and an aluminum bar a diameter of 2.49 in. Figure 6 shows the large difference in length (or volume) between a titanium rod and a steel rod of equal weight.

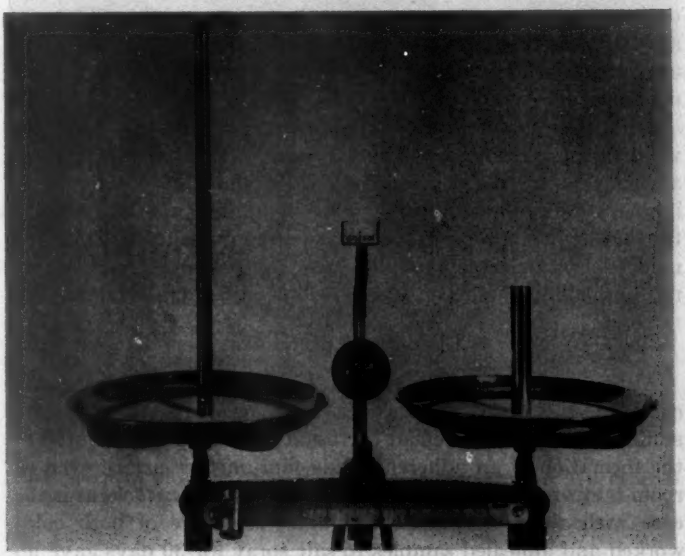


FIGURE 6.
Large difference in size between rod of Titanium (left balance pan) and rod of steel (right balance ban) of equal weight.
U. S. Bureau of Mines photo.

It can be readily observed from an examination of the values shown in Table III, that the properties of titanium compare most favorably with other well known commercial metals and alloys in both the annealed and cold-worked state. Since yield strength is often a more important criterion for serviceability than the ultimate tensile strength, it will be used here as the basis for comparison. The ratio of yield strength to density is equal to or

*0.2% Offset.

greater than most of the well known structural metals and closely approaches the value of a quenched and tempered alloy steel with a yield strength of 250,000 lb/sq in. It should be remembered that the values shown for the aluminum alloys 24ST and 75ST represent the highest strength alloys perfected by the aluminum industry. The values for the magnesium alloy AM59S are also representative of very high-strength magnesium alloys. The upper range of properties listed for the alloy steels is not typical of usual structural steel materials but is achieved by a high degree of alloying and optimum heat treatments. Thus, it appears that on a strength-weight basis titanium may well find application where highly alloyed steels and light metals are conventionally employed.

It must be borne in mind that the above comparison is only valid for pure titanium. There is considerable effort underway at this time for the development of titanium-rich alloys. Preliminary data available thus far bear out the expectations that the mechanical properties of pure titanium will be enhanced manyfold by suitable alloying. Similar to steel, certain of the titanium alloy systems appear to be amenable to heat treatment.

Corrosion Resistance.

Although the metal titanium has high chemical activity, the corrosion resistance is, however, comparable to that of 18-8 stainless steel. Its excellent corrosion resistance is attributed to the formation of an adherent oxide film on the surface even at room temperature which protects it from further attack, as is the case with aluminum and chromium. Exposure of both cold-worked and annealed samples to salt spray for 30 days gave no visible sign of attack, the tensile properties of the exposed samples being the same as control samples. Figure 7 shows the results of an outdoor exposure test sufficient to badly rust a steel sample, whereas the two titanium specimens are bright and unattacked. Concentrated nitric acid, dilute nitric acid, hydrochloric acid, acetic acid, ammonium and sodium hydroxide do not appreciably attack titanium. Concentrated sulfuric and hydrochloric acids however rapidly attack the metal.

At elevated temperatures, titanium undergoes surface oxidation and still more detrimentally reacts with atmospheric gases and becomes brittle. However, as in the case of probable

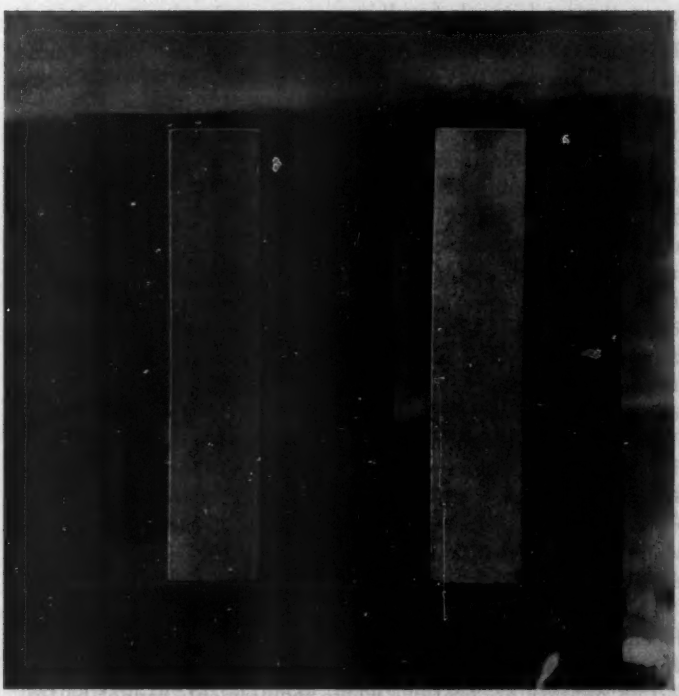


FIGURE 7.
Steel specimen in center badly rusted after outdoor exposure test, whereas two Titanium specimens remained bright and unattacked.
U. S. Bureau of Mines photo.

improvement of its good mechanical properties, there seems to be no reason not to expect even greater oxidation and thermal resistance of certain titanium-rich alloys.

POSSIBLE USES AND APPLICATIONS.

The properties of titanium, even in the present state of exploitation, indicate its extensive potentialities for application to both military and civilian usage. Its outstanding characteristics of lightweight, high strength and corrosion resistance make it a most desirable structural metal. Its high proportional limit and its low density make it a preferred structural material where

minimum weight and high stresses are design requirements. This immediately indicates its possible use an an air-frame structural metal. Its excellent corrosion resistance to salt spray indicates its utility for numerous shipboard applications and for pontoons and other similar aircraft components where the requirement for inertness to sea water corrosion is essential. The ability of titanium to be surface hardened suggests its use for parts which are subject to frictional wear. Pistons, cylinders and other engine components seem likely applications. The high proportional limit and low modulus of elasticity make titanium a useful material for springs where considerable extension with low loads is required. Titanium has a high electrical resistance and may be useful as a metallic electrical resistor. Its unusual property of marking glass is useful in the production of stable high electrical resistances.

Because of its high melting point, there is some hope that titanium and its alloys will prove to be a useful heat-resistant material. However, at the present writing, alloy development work and high temperature tests have not reached the stage where definite conclusions may be drawn. This field of research is being vigorously attacked.

The large scale applications of titanium and its alloys will depend upon their availability in suitable form and at a cost where they will not be considered as precious metals. For certain military applications where requirements cannot be fulfilled by other materials, even the present price does not seem exorbitant. The Bureau of Mines is presently producing titanium at about $3.00/lb. and it is estimated that large scale production may bring the price down to 0.50 to $1.00/lb. The history of aluminum and magnesium and the tremendous reduction in price levels of those two metals should be borne in mind when considering the future of titanium research.

As indicated previously, the major reason for the lack of commercial exploitation of titanium was not due to the scarcity of its ores but to the brittle characteristics of the metal as prepared by the early investigators. This apparant "inherent" brittleness was subsequently shown to be caused by the presence of gases (e.g., hydrogen, oxygen, nitrogen) or other elements in the metal. A reaction between these gases and titanium often results in hard but severely embrittled alloys. In addition to the ready absorp-

tion of gases, titanium reacts rapidly at elevated temperatures with many metals and practically all crucible materials including alumina, beryllia, and carbon, particularly in the liquid state. This high degree of reactivity creates a source of impurities and poses difficult problems for casting the metals by conventional methods. It is necessary to use vacuum techniques, inert atmospheres, or other protective measures for handling titanium at elevated temperatures, both in production and fabrication processes.

Government Programs.

Numerous government programs conducted either in the laboratories or under the sponsorship of the Navy, Bureau of Mines, Army and Air Force are directed toward increasing the availability of titanium (and consequent reduction of price); development of techniques and methods for the production of pure, ductile titanium; fundamental research on titanium and its alloys; and development of titanium alloys. The Navy is making a major contribution to the exploitation of this important metal.

Recognition should be given to the Bureau of Mines for its initiative and foresight in developing ductile titanium and in continuing research in this field. In addition to operating the pilot plant at Boulder City, Nevada, for the production of titanium powder, the Bureau of Mines is studying various beneficiation methods for the extraction of the ore and production methods for preparation of the pure metal. The major factor leading to the present high cost of titanium is the cost of titanium tetrachloride at $1.04/lb., whereas the ilmenite ore only costs 0.04/lb. Thus, a revision of present production procedure can result in a large price reduction. The Bureau of Mines is also planning to make an exhaustive study shortly of the titanium reserves of this country and the exploitation of its ores.

Titanium powder produced by the Bureau of Mines' process is fabricated into shape by powder metallurgy techniques. It would be more desirable for production and other purposes if the metal could be cast and then fabricated by conventional forging and rolling methods. A large part of the current effort in the field of titanium is aimed at finding satisfactory crucible materials and developing new casting methods.

To date some success has been achieved by vacuum melting in

a graphite crucible using resistance or induction heating. Melts have been made which could be readily forged and hot worked despite some carbon contamination. Improvements of this technique are being explored and the search goes on for even more satisfactory crucible materials.

Another casting process that offers considerable promise is the arc casting process (vacuum or atmospheric). Alloys with extremely high melting points can be melted and cast by this process without the ensuing crucible problems of conventional casting methods. The process was successful for the casting of molybdenum and results to date indicate that arc-casting methods can also be successfully applied to the production of ductile titanium and titanium alloys. Quite sizeable ingots have been prepared by this method. The arc-casting process makes use of ingots or bars which were originally pressed from titanium powder made by the magnesium-titanium tetrachloride reduction process or rods made by the Van Arkel titanium tetra-iodide decomposition process. Essentially the process consists of using such a bar of titanium as one electrode (or titanium powder may be fed through a hollow electrode into the arc), the other electrode being a layer of titanium at the bottom of a water cooled copper crucible. The arc is struck in an inert atmosphere and the molten titanium formed from the consumable electrode falls into the cooled copper container and is rapidly solidified there.

In conjunction with the development of new casting methods for the production of ductile titanium and its alloys, considerable effort is being expended on the improvement of powder metallurgy techniques for compacting the titanium powder into forgeable ingots. Vacuum sintering or sintering in dry hydrogen and cooling in vacuum seem to yield promising results. An important aspect of the above research program will be the comparison of properties of the metals as made by powder metallurgy, conventional casting, or arc-casting processes.

As important as the research aimed at the production and fabrication of ductile titanium are the numerous programs concentrating on the development of titanium base alloys. As indicated previously, this is a relatively unexplored field and one which exhibits promising potentialities. At the present time, titanium-rich systems of iron, cobalt, nickel, chromium, copper, tungsten, manganese, aluminum, silicon, boron, vanadium,

beryllium, zirconium, molybdenum, carbon and oxygen are being studied and the physical, electrical, mechanical, metallographic, chemical and fabricating properties of promising alloy systems are being determined. Titanium seems to be readily weldable and methods and techniques for satisfactory welding procedures are also being studied. Corrosion studies of titanium and its alloys occupy an important position in the program and salt-water corrosion, corrosion-fatigue, stress-corrosion and erosion-corrosion fatigues tests are underway. Several sheets of titanium have been sent to Kure Beach, North Carolina, for atmospheric corrosion testing.

Equilibrium (constitution) diagrams of many of the above titanium-rich binary alloys will be constructed and the effects of added elements on phase transformations and properties of titanium accurately measured. In view of the importance and potent effects of small amounts of oxygen, the titanium-oxygen system is being studied. A proper evaluation of other alloy systems can only be made after the effects of small amount of oxygen and other gases are known, so that a proper base line can be drawn. This project necessitates the development of exact analytical procedures for the accurate determination of extremely minute amounts of oxygen.

The evaluation of the high temperature properties of titanium and its alloys is also currently receiving much attention. Theoretical considerations have indicated the potentialities of several alloy systems for high-temperature materials, but as stated previously, insufficient test data are available for drawing definite conclusions.

It is interesting to note that the Naval Air Material Center laboratories (Naval Air Experimental Station, Philadelphia, Pa.) of the Bureau of Aeronautics is engaged in the design, fabrication and evaluation of a titanium alloy float. This is regarded as an experimental evaluation of the material in a practical naval application.

Industrial Research.

There is also considerable industrial activity aimed at the research and development of titanium and its alloys. Several major prominent corporations are engaged in this field and commercial exploitation is a distinct probability for the not too distant future.

Symposium on Titanium.

In view of the extensive interest and activity of both government and industry in titanium, the Office of Naval Research is planning to sponsor a "Symposium on Titanium" in the near future. Both industrial concerns and government agencies will be asked to participate and review their current research programs. The titanium research effort is constantly expanding. It, therefore, seems opportune at this time to evaluate the results of the work done to date in the light of our needs and to determine appropriate paths for future research.

Conclusion.

It is thus apparent that as new needs have arisen for metals which are capable of withstanding the severe requirements of modern military technology, metallurgists have been compelled to turn to new sources to solve the complex materials problems imposed by these requirements. A potentially fruitful source of new materials for the solutions to many of these problems seems to be the group of metals known as the "Refractory Metals." One of this group, titanium, is rapidly achieving an important status in metallurgical circles and research. It is anticipated that the large number of current research and development programs will develop the potentialities of titanium which will then justify the name bestowed upon it, "Titanium—the Titan of Metals."

Much of the information presented in the above discussion was obtained from Bureau of Mines' publications listed in the references which follow. The reader is referred to these articles for more details concerning the history, properties and production of titanium and its alloys. All of the illustrative figures were supplied through the courtesy of the Bureau of Mines.

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PROPELLER CHARACTERISTICS UNDER NON-AXIAL FLOW.

By Commander C. R. Hirschberger, U.S.N.
and
Commander Frank C. Jones, U.S.N.

The photograph on the following page demonstrates that cavitation under non-axial flow varies with blade position and does not remain the same as the blade make its complete revolution. Therefore, in addition to the generally accepted conditions, the degree of cavitation depends on blade position when the entering flow-lines are not parallel to the propeller axis.*

As Wilde put it, "Experience is the name everyone gives to his mistakes." In setting up propeller tunnel conditions for a cavitation run for determining thrust and torque coefficients under non-axial flow, the stroboscope was inadvertently left energized. This mistake, or happy coincidence if you will, resulted in the observation of the varying-degree of-cavitation phenomenon.

The investigation undertaken as a result of this observation was the subject of a thesis prepared by the authors at Massachusetts Institute of Technology (1). The investi-

The Authors.

Commander C. R. Hirschberger graduated from the USNA in 1937. Various duties aboard the U.S.S. *Mississippi* from 1937 to 1940. April 1943 received Master's Degree in Naval Engineering and Construction after three years of graduate work at M. I. T. Philadelphia Navy Yard from April 1943 to April 1944. Chief Engineer, U.S.S. *Boise*, in the South Pacific from May 1944 until the end of the war. October 1945 to Spetember 1947, Charleston Naval Shipyard as Design Superintendent and Assistant Planning and Estimating Superintendent. Since October 1947 has been in charge of the Gas Turbine, Mechanical and Bearings Laboratory at the U. S. Naval Engineering Experiment Station, Annapolis, Md.

Commander Frank C. Jones, U.S.N., graduated from the Naval Academy in the Class of 1938, and after two years sea duty, took a three year course in Naval architecture and engineering at M. I. T. graduating with the degree of Master of Science in that field. He was subsequently in 1943 designated for Engineering Duty. Two years duty in the Puget Sound Navy Yard in connection with repair and overhaul of ships was followed by similar duties at the Naval Operating Base, Guam, until 1947. Commander Jones has been attached to the Bureau of Ships, Design Division, since August 1947.

*NOTE: This information was not previously released as it had been classified "Restricted" by the Navy Department.

PHOTOGRAPH A: The water is moving from right to left, ascending at an angle of 7° to the propeller shaft. The upper blades are moving to the left.

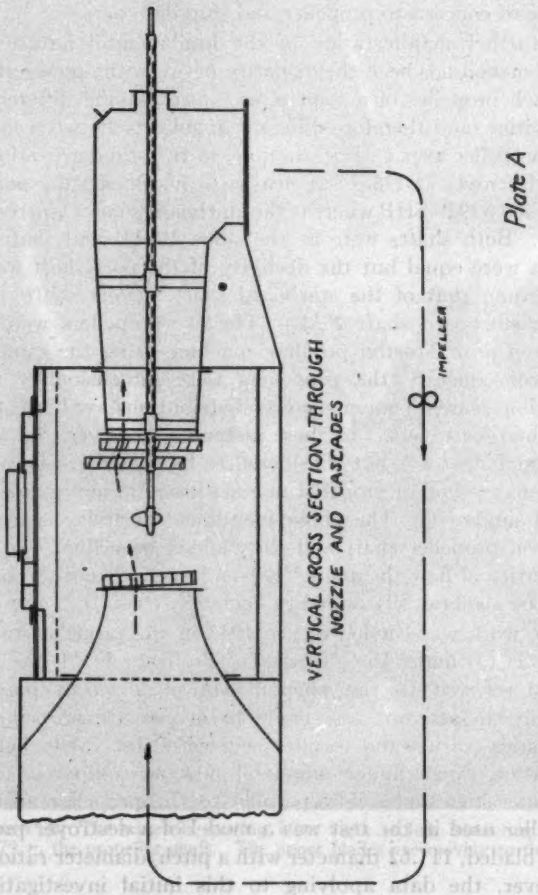
gation explains qualitatively and at least in part the reason for the discrepancy between the start of cavitation under full-scale operating conditions and that of a model propeller as heretofore tested in a propeller tunnel under axial flow. The results of the investigation show that cavitation occurs earlier (at lower slip) under non-axial flow. Inasmuch as propeller characteristics are often determined from propeller tunnel tests the discrepancy between predicted and actual starts of cavitation has been a source of concern to propeller and ship designers.

A further manifestation of the fundamental nature of this phenomenon has been the disparity between the power absorbed by each propeller of a twin screw vessel having different shaft declivities (and therefore different angularity between flow lines and propeller axes.) For example, in the standardization trials of a destroyer (DD837) at heavy displacement, the port shaft absorbed 32920 SHP whereas the starboard shaft absorbed 34470 SHP. Both shafts were at the same RPM. and shaft spread angles were equal but the declivity of the port shaft was 1°30′ more than that of the starboard shaft. (Port shaft declivity 4°04′; starboard shaft 2°34′). The two propellers were manufactured on a Morton profiling machine using the same model and consequently the propellers were dimensionally similar. Additional cases of unequal power distribution have been observed on other destroyers. In these instances, however, the effect of non-axial flow was not considered to be conclusive because the previous method of propeller manufacture did not assure dimensional similarity. The above-mentioned declivity is the angle between propeller shaft and the molded base line. To obtain angularity of flow the angle between flow lines and the base line must be algebraically added to declivity.

The work was carried out in 1943 in the propeller tunnel at M. I. T. (2) under the guidance of Professor F. M. Lewis, who briefly referred to the phenomenon in a recent paper (4). Initially an attempt was made to investigate propeller characteristics (thrust and torque coefficients) for values below the cavitation range under non-axial flow as compared to these characteristics under flow parallel to the propeller axis. The propeller used in the test was a model of a destroyer propeller, three bladed, 11″.62 diameter with a pitch/diameter ratio of 1.1. However, the data applying to this initial investigation are

insufficient and considered unreliable due to the rapid and inconsistent changes in the comparison ratios. The data do indicate a comparatively small variation in both torque and thrust coefficients below the cavitation range under angularity of flow up to 12°.

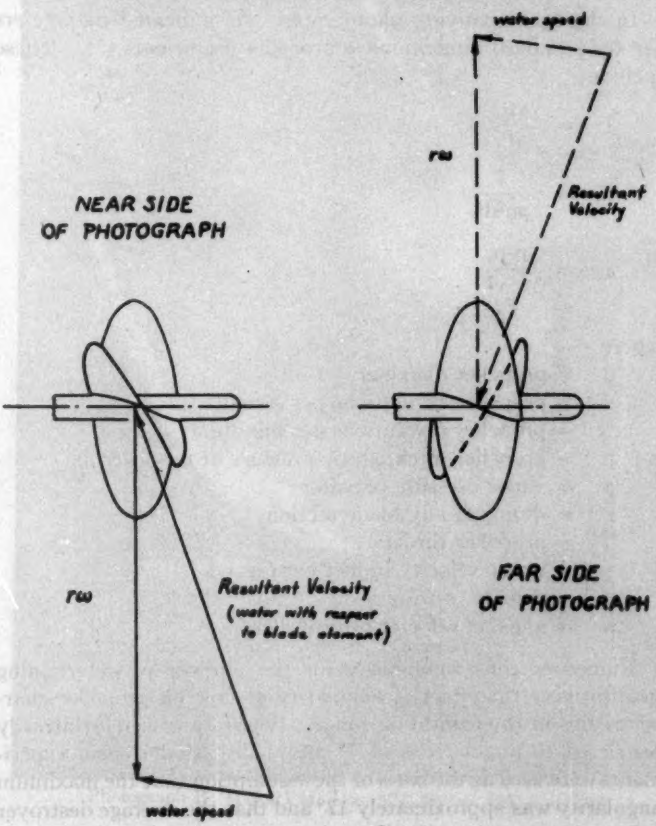
Angularity of flow in the tunnel was achieved by cascades as indicated in the below sketch.



After the observation of the varying degrees of cavitation with blade position, the investigation was diverted to an analysis of this cavitation phenomenon. It was determined that the reason for the varying degree of cavitation as the blade rotated was the varying angle of attack between the blade section and the entering flow lines.

PLATE B

DIAGRAMMATIC DELINEATION OF PHOTOGRAPHS



As can be seen from the above sketch, the vector addition of the blade rotative speed ($r\omega$) and the water speed results in varying angles of attack of the blade sections to the entering flow lines. This varying angle of attack as the blade rotates is the reason for the varying degree of cavitation. It is further pointed out, as demonstrated by the photographs, that the degree of cavitation is a function of the instantaneous angle of attack. The formation and cessation of cavitation occurred sufficiently rapidly to be clearly discernible in the photographs despite model propeller speeds ranging from 1125 RPM. to 1620 RPM.

In the accompanying photographs, the indicated parameters are the standard dimensionless propeller coefficients (3). These include:

$$J = \frac{vn}{d}$$

$$K_T = \frac{T}{\rho n^2 d^4}$$

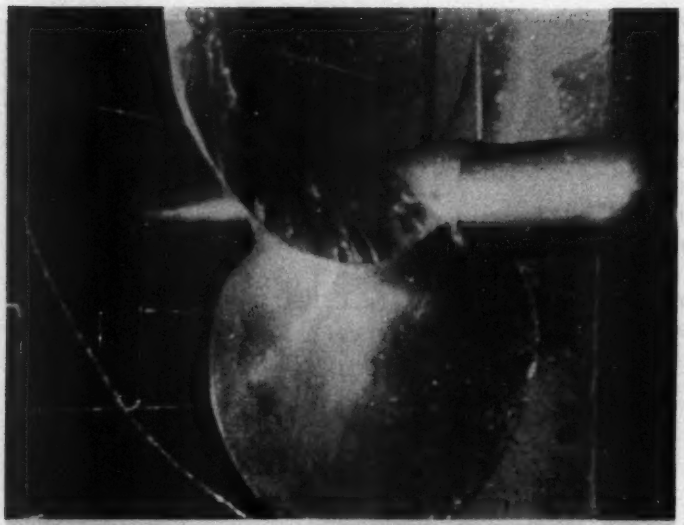
$$\sigma = \frac{p - p_v}{\frac{\rho v^2}{2}}$$

where

- d = propeller diameter
- n = propeller revolutions per second
- N = propeller revolutions per minute
- p = propeller pitch; static pressure at propeller axis
- p_v = vapor pressure of water
- r = radius at any blade section
- T = propeller thrust
- v = water velocity in feet per second
- ρ = density of water
- ω = angular velocity of propeller

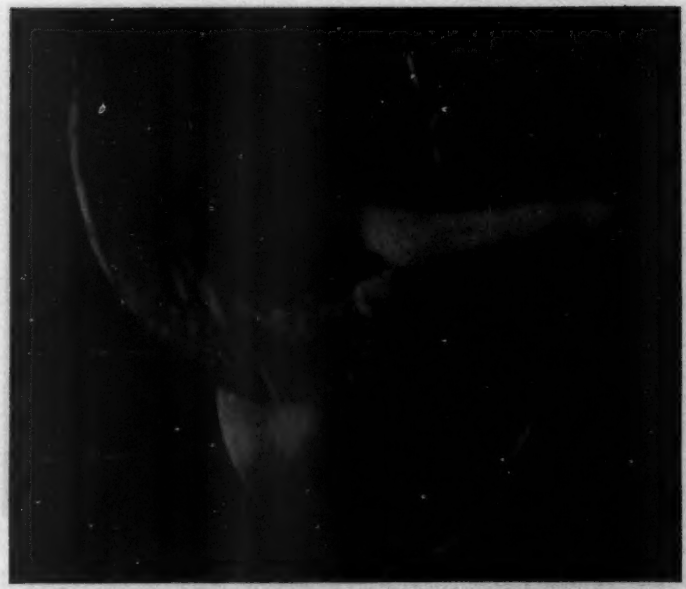
Numerous runs were made for the purpose of determining qualitatively the effect of angularity of flow on propeller characteristics in the cavitation range. These runs were arbitrarily restricted to angularities of 7° and 12°. Model basin experiments were used as the basis of the assumption that the maximum angularity was approximately 12° and that the average destroyer value was approximately 7°.

Photograph B: 1125 RPM, J = .974, K_T = .122, σ = 1.45



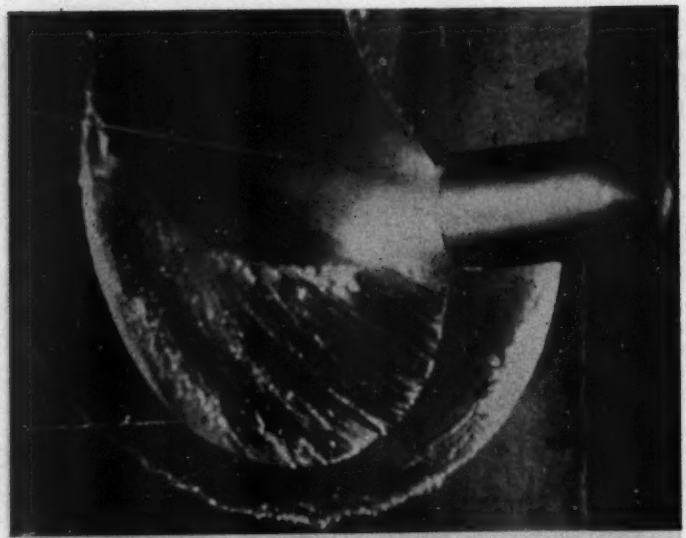
Photograph 1: 1145 RPM, J = .966, K_T = .107, σ = 1.43

Photograph 2: 1145 RPM, J = .966, K_T = .107, σ = 1.43



Photograph 3: 1145 RPM, J = .966, K_T = .107, σ = 1.43

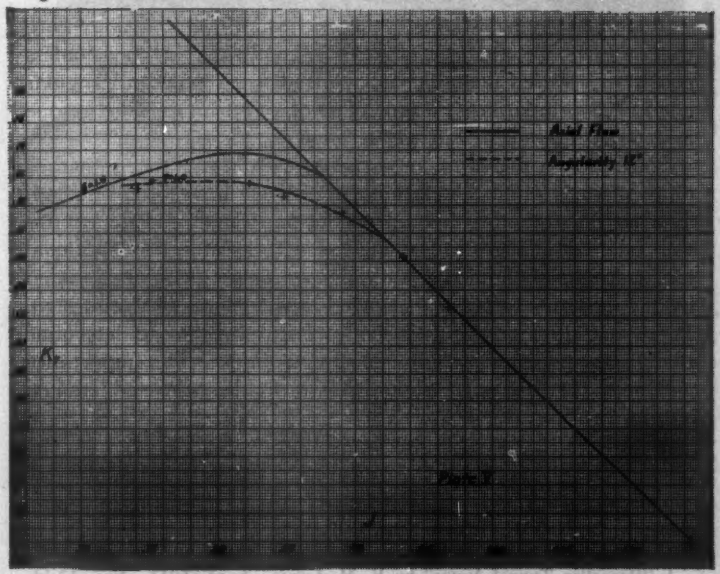
Photograph 4: 1190 RPM, J = .87 , K_T = .154, σ = 1.65

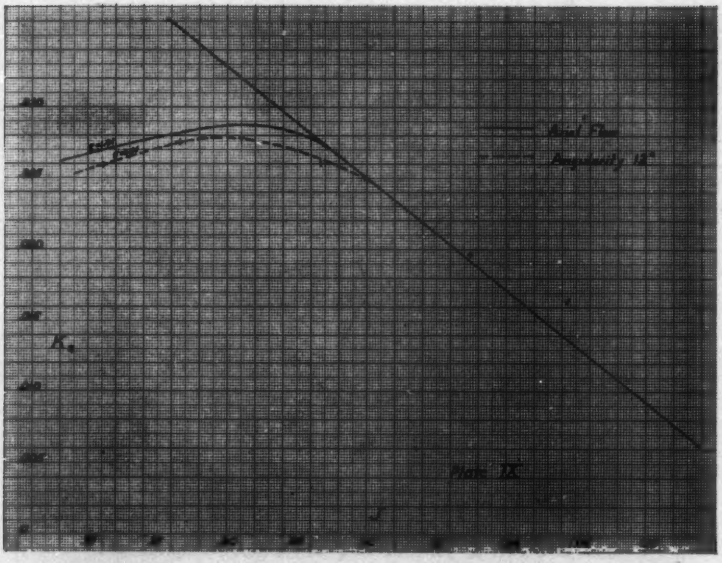


Photograph 5: 1190 RPM, J = .87 , K_T = .154, σ = 1.65

Photograph 6: 1190 RPM J = .433, K_T = .291

Photograph 7: 1620 RPM, J = .672, K_T = .142





Runs at varying values of the cavitation index, σ, were conducted both for the 7° and the 12° condition. Dimensionless coefficients were calculated and plotted, with axial and angular conditions compared. Sample plots of a 12° run for both torque and thrust coefficients are given below.

Since this paper is of an explanatory nature, the numerous data, plots, sample calculations, etc. of (1) have been omitted. The detrimental effect upon propeller performance in the cavitation range of changes in the angle of attack caused by non-axial flow is a new phenomenon and apparently not fully appreciated by the majority of persons engaged in propeller and ship design. It remains therefore for additional work to be done to determine the effects quantitatively for design purposes. The importance and need for that work should not be underestimated, nor its accomplishment delayed.

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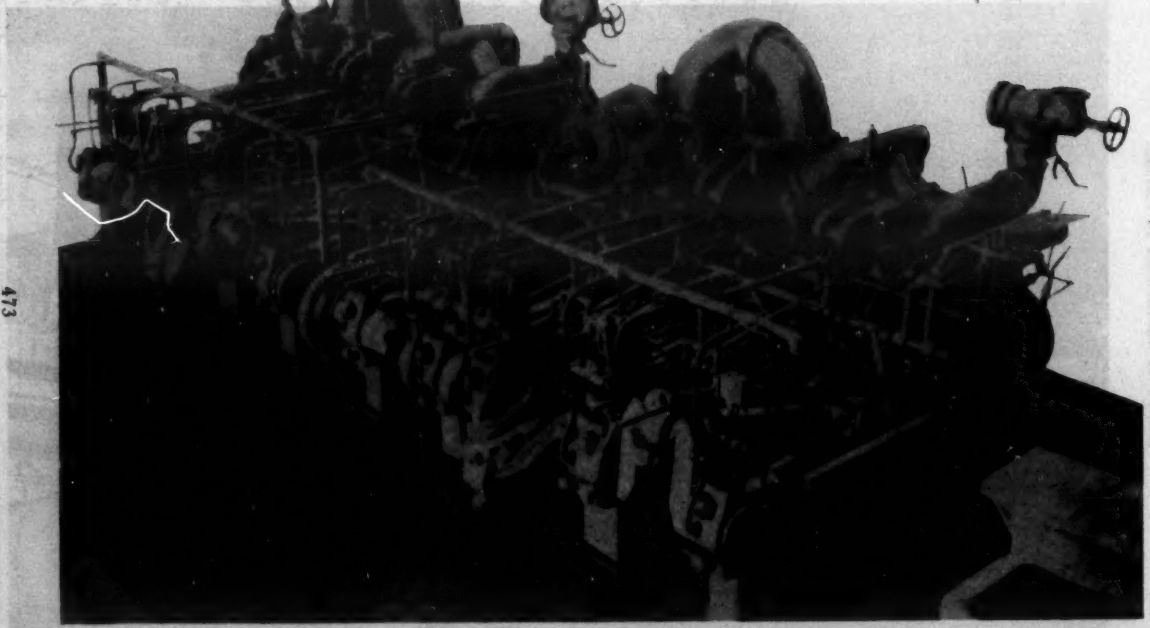
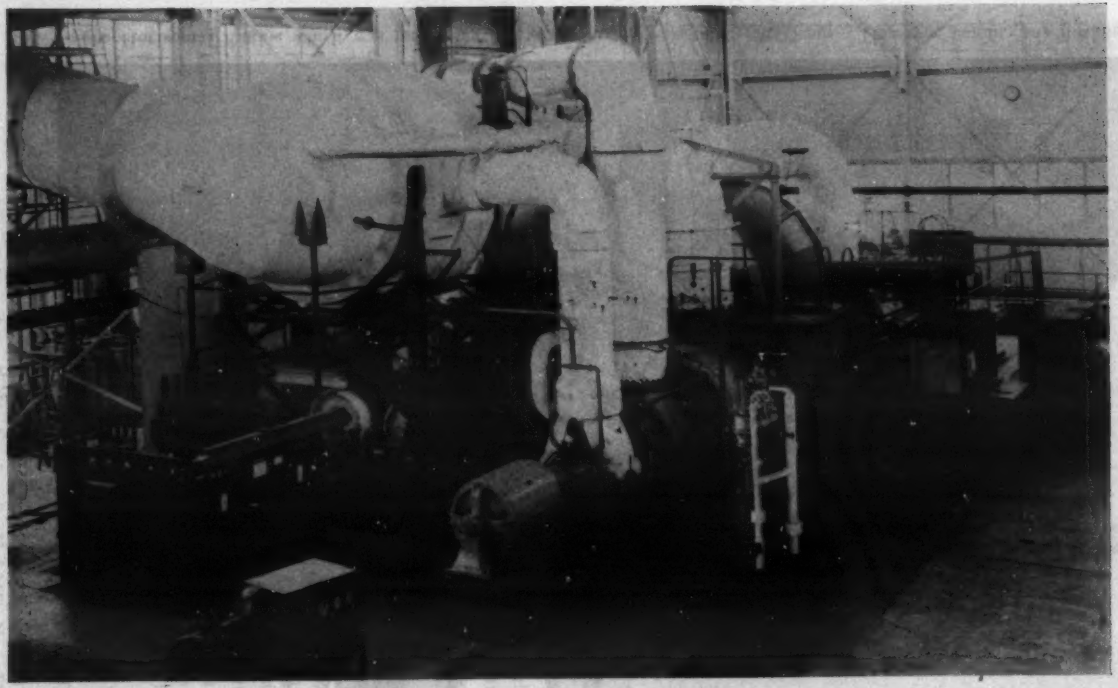


The Horizontal Engine which was just going out when the Society came into being. This type engine was installed in the *Baltimore*.



TODAY'S EXPERIMENTAL GAS TURBINE.

Not a seagoing unit but possibly a forerunner of one. A far cry from the horizontal engine of the era of the Society's birth.

EFFECTS OF INCLUSIONS ON THE ENDURANCE PROPERTIES OF STEELS.

By William C. Stewart
and
W. Lee Williams.

Introduction.

Navy Department Specifications 49S2 and 46S39 for steel forgings, as well as other specifications for steel products, contain a clause that materials shall be free from excessive non-metallic inclusions and segregation. Experience in the field has shown the need for defining the word "excessive" as it applies in this case. Since all steel contains inclusions, it is necessary to establish levels for acceptability depending on a number of factors such as the purpose of the forging, its size, the distribution and form of the non-metallics and the presence of other defects.

Ordinarily, Inspectors of Naval Material experience their greatest difficulties when decisions must be made as to whether or not inclusions are excessive in large forgings. Not so much is at stake in the inspection of small parts. Neither does much doubt

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Mr. W. Lee Williams received the degree of Bachelor of Science in Chemical Engineering in 1941 from Antioch College. During four of his student years he served under the cooperative plan as laboratorian in Physical Metallurgy at Battelle Memorial Institute. After graduation he became Metallurgist for the American Steel Abrasives Company. During the war he was commissioned in the Naval Reserve and served at the U. S. Naval Engineering Experiment Station. After discharge, he remained as a civilian metallurgist and became Assistant to the Superintendent of the Metallurgical Laboratory.

exist in passing or rejecting small parts to be highly stressed in engines or aircraft, and manufacturers usually concede that such parts must be made as near to perfection as possible. On the other hand, a large forging, such as a 14 inch propulsion shaft, is not likely to be stressed so highly. Consequently, an Inspector is gravely concerned as to whether or not large non-metallic inclusions are harmful, particularly in view of the major expenditures of funds and materials which are involved and of possible delays to scheduled production programs.

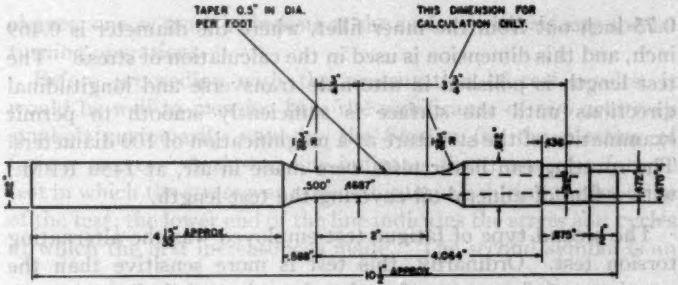
In this connection, a large number of tests have been conducted at the U. S. Naval Engineering Experiment Station on materials rejected because of so-called "excessive" inclusions. Most of the tests were made with specimens removed from heavy forgings, and the principal interest was directed toward an estimation of the serviceability of large machinery components containing unusual amounts of non-metallics.

A review of the work indicated that the inclusions could be classified into a few general types, depending on structure and distribution, and that each type led to the same general results and conclusions. Consequently, it will be a purpose of this paper to present the results of only a few tests, these representing typical cases for each type of inclusion. After the tests have been described, an attempt will be made to evaluate the results; this part of the paper will contain some theoretical considerations on the behavior of inclusions as stress concentrators.

Method of Test.

In nearly all cases, the presence of large amounts of non-metallic inclusions does not alter, to any significant extent, the static properties of large machinery components. The principal danger from such inclusions lies in the possible lowering of the fatigue strength as a result of stress concentration in the metal adjacent to the inclusions. It is logical, therefore, that the primary concern is the fatigue testing of specimens cut from inclusion bearing steel.

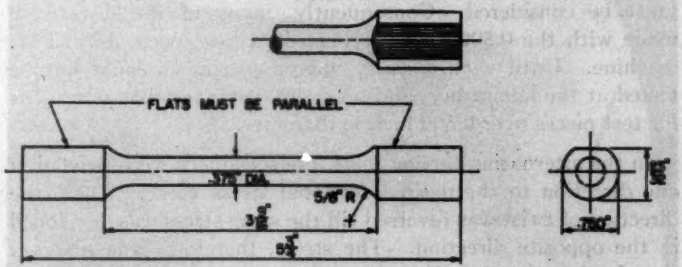
Two general types of fatigue tests have been employed. The first is the White-Souther rotating cantilever test using the specimen shown in Figure 1. The gauge length of this specimen is conically tapered so that the stress does not vary over 1.5 per cent over a length of 1.5 inches. The maximum stress occurs



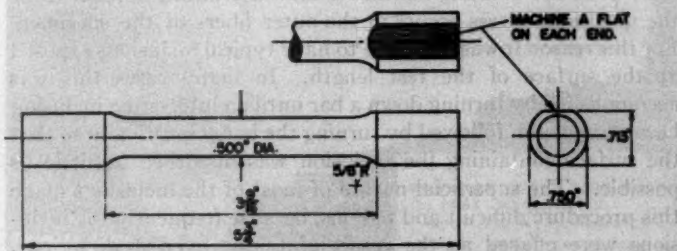
SPECIMEN TYPE Cc

ROTATING CANTILEVER ENDURANCE SPECIMEN AS USED BY U.S. NAVAL ENGINEERING EXPERIMENT STATION.

FIGURE 1.



MODIFIED SPECIMENS FOR HIGH SPEED ALTERNATING TORSION MACHINE.



STANDARD ALTERNATING TORSION SPECIMEN.

FIGURE 2.

0.75 inch out from the inner fillet, where the diameter is 0.469 inch, and this dimension is used in the calculation of stress. The test length is polished in alternate transverse and longitudinal directions until the surface is sufficiently smooth to permit examination of the structure at a magnification of 100 diameters. The rotating cantilever tests were made in air, at 1450 RPM., with a film of mineral oil covering the test length.

The second type of fatigue test employed was the alternating torsion test. Ordinarily, this test is more sensitive than the rotating cantilever test for the detection of fatigue damage caused by inclusions, a point to be elaborated later. In Figure 2 are shown the two specimens which have been used in these tests. The 0.375 inch specimen was run at 1450 CPM. in a high speed machine. This specimen is rather small where size effects must be considered. Consequently, many of the tests were made with the 0.500 inch specimen in a low speed, 360 CPM. machine. Until very recently, larger specimens could not be tested at the laboratory, and no results are available at present for test pieces over 0.500 inch in diameter.

In the alternating torsion tests, the specimens were twisted in one direction to the desired torsional stress value. Then, the direction of twist was reversed till the same stress was developed in the opposite direction. The stress, therefore, was reversed completely on each half cycle, and stress-cycle curves could be established in torsion in the same way that ordinary flexural fatigue curves are obtained.

In both the rotating cantilever and alternating torsion tests, the maximum stress occurs in the outer fibers of the specimen. For this reason it was desirable to have typical inclusions exposed at the surface of the test length. In many cases this was accomplished by turning down a bar until an interesting inclusion became exposed, followed by turning the bar eccentrically so that the surface containing the inclusion was disturbed as little as possible. The superficial nature of most of the inclusions made this procedure difficult and tedious, because frequently the inclusions were effaced as the result of taking even a slight cut. Hence, the desired locating of inclusions could not be accomplished in many cases. Usually, however, the number of inclusions within a small area was great enough to expose, through

chance, one or more inclusions at the surface without any special turning operations.

Before proceeding with the presentation of test results, it would be well to mention here the significance of two arbitrary symbols customarily used by the Station for the plotting of fatigue curves. A zig-zag line indicates the course of a fatigue test in which the stress was raised at intervals during the progress of the test; the lower end of the line indicates the stress and cycles at which the first increase was made. The second symbol is an

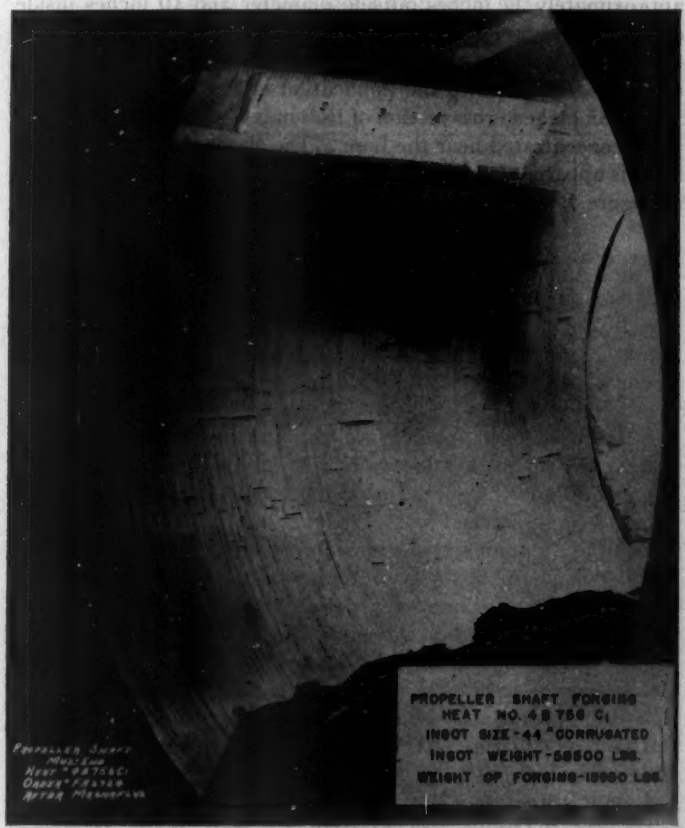


FIGURE 3.

arrow placed on a test point; this indicates that a specimen has been removed at this point without failure.

Results of Tests on Typical Cases of Inclusion Bearing Steel.

Longitudinal Inclusions With Random Distribution. The first type of inclusions to be reported herein might well be classified as longitudinal with random distribution. An excellent example of this type was found in a propulsion shaft which measured approximately 17 inches outside diameter and 10 inches inside diameter. Figure 3 illustrates the appearance of the inside surface after magnaflux inspection; however, most of these inclusions could be seen prior to the application of the magnetic particle test. An etched cross section of the shaft revealed the inclusions to be concentrated near the bore. The slag nature of the inclusions is apparent from the unetched longitudinal structure shown in Figure 4.

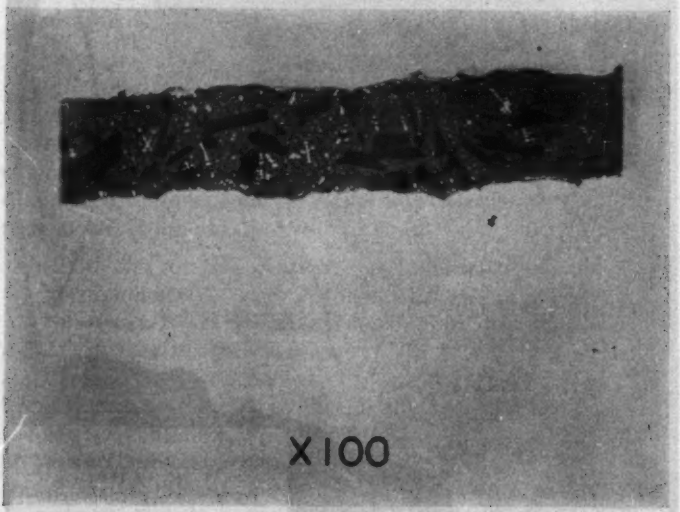


Figure 4.

The average composition of the shaft was that of a molybdenum-vanadium steel, as follows:

C.....0.32 %	Si.....0.27%
S.....0.020	Ni.....0.05
P.....0.014	Mo....0.46
Mn....0.62	V.....0.20

Due to the localized distribution of inclusions (near the bore), it was decided to determine the longitudinal tensile properties for three areas from the outside to inside of the forging. Results are given in Table I.

TABLE I
LONGITUDINAL TENSILE PROPERTIES OF
17" O. D. Mo-V SHAFT.

Property	Actual Shaft			Spec. 49S2 (INT) Class AN
	Outside	Midway	Inside	
Tensile Str.—PSI.....	80500	77500	76400	80000
Proof Stress—PSI.....	47600	47300	47600	45000
Yield Point—PSI.....	48800	48200	48100	.....
Elong. in 2"—%.....	31.6	31.5	33.0	25.0
Red. of Area—%.....	60.0	62.6	62.8	45.0

No rotating cantilever fatigue tests were made on material from this shaft. At the time the test was conducted, it was known from previous experiences that the rotating cantilever properties would not be affected by this type of inclusions. More will be said on this point later on. The fatigue tests were limited to the alternating torsion method, using both sizes of specimens shown in Figure 2. The results of the tests are represented by the three curves shown in Figure 5. The broken line curve with solid points represents a few tests made in the high speed machines at 1450 stress reversals per minute. The solid lines with open points represent tests made in the low speed machines at 360 stress reversals per minute. The specimens were taken from the outside, midway and bore sections of the shaft, as indicated by the symbols for the test points.

It can be noted that the greatest scattering of test points occurred with the specimens taken near the bore of the shaft where heavy inclusions were located. The very fact that scatter is increased is, in itself, an indication that the inclusions have reduced the fatigue resistance of the steel. Actually, there is a spread of about 5000 psi between the estimated endurance

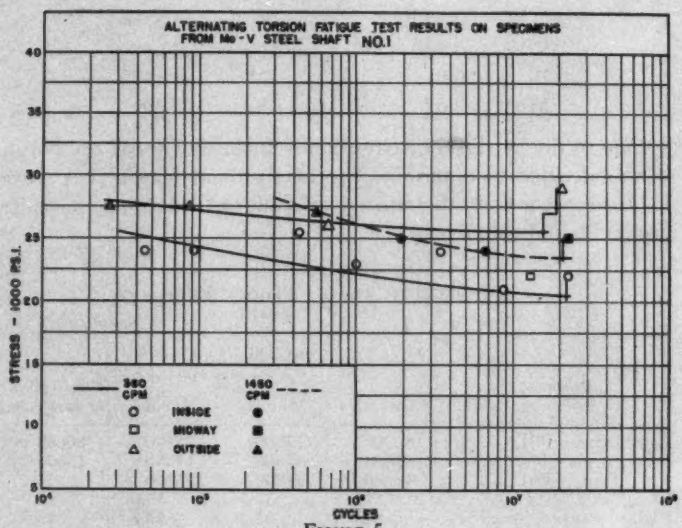


FIGURE 5.

limits as obtained for the outside and inside specimens. The endurance limit obtained in the high speed tests is about midway between the curves for the low speed tests.

The average overall tensile strength of the forging is about 78,100 psi. The endurance ratio (ratio of endurance limit to tensile strength) then would be 25500/78100 or 0.33 for the specimens near the outside and 20500/78100 or 0.26 for the specimens in the region of heavy inclusions. The value of 0.26 is not unusual for many steels of similar structure and strength, although the fact remains that in this particular steel the fatigue strength in the bore area was reduced nearly 20% by the presence of large non-metallics.

It must be kept in mind that the fatigue tests were made on small specimens. Furthermore, the damaging inclusions occurred near the bore of the propulsion shaft where service stresses are low. The significance of these facts, as they affect the usefulness of the shaft in service, will be considered later in the report.

Another example of longitudinal inclusions with random distribution was found in the breech end journal of a large forged

rotor. The smallest section on the rotor was 6 inches in diameter. The inclusions were essentially straight and longitudinal, and in general were larger than the ones found in the propulsion shaft described previously. However, a few of the inclusions, which were exposed on the surfaces of the test specimens, varied a few degrees from the longitudinal, and this apparently caused the alternating torsion properties to be slightly lower than usual in some cases. The inclusions had a maximum length of about 0.8 inches and did not occur generally throughout the section.

The composition and properties of this forging were found to be as follows, with the tensile specimens cut from an area free of large inclusions. The material was an AN nickel type steel.

C.....0.30 %	Si.....0.21%
S.....0.017	Cr.....0.25
P.....0.045	Ni.....2.65
Mn....0.67	Mo....0.40

Tensile Strength	96,800 psi
Yield Strength, .1% Set	75,000 psi
Proof Stress	74,500 psi
Elong. in 2"	26.0 %
Reduction of Area	61.2 %
Brinell Hardness	215

Due to the fact that some of these inclusions were slightly off from the longitudinal, it was decided to make rotating cantilever flexural fatigue tests on this steel. The results of such tests are presented graphically in Figure 6. Very little scatter of test points occurred, and an endurance limit of 48,000 psi is indicated for inclusion bearing specimens. The endurance ratio is 48000/-96800 or 0.495. This ratio is normal, and even might be considered high for material of this composition and structure. Hence, it must be concluded that the inclusions, even though slightly off from longitudinal in some cases, did not effect the flexural fatigue resistance.

Alternating torsion tests, using 0.500 inch diameter specimens, also were made on this material. The results are presented in Figure 7. The varying nature from one inclusion to another caused considerable scatter in test points; hence, it was not possible in this case to draw even a representative stress-cycle

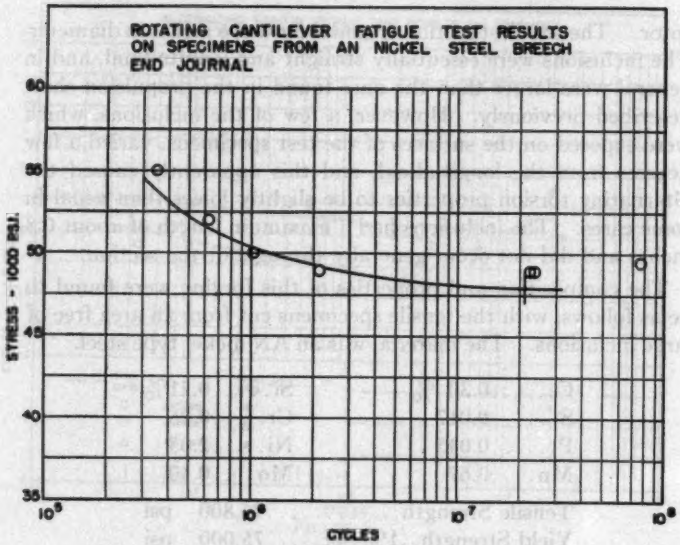


FIGURE 6.

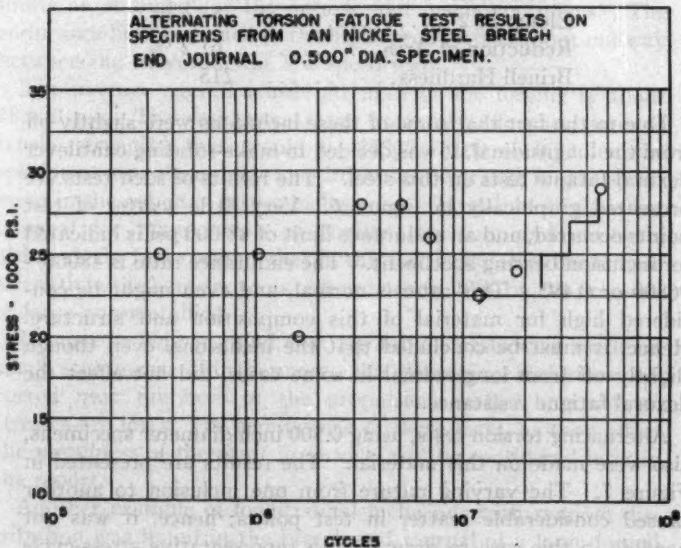


FIGURE 7.

curve. In general, it was noted that lower values were found for specimens with more and larger inclusions. One of the specimens broke at 1,660,000 cycles while running at a stress of 20,000 psi. All other specimens were superior to this, and hence it is probably safe to assume that 20,000 psi would represent about the lowest fatigue strength of the inclusion bearing metal. Such a value would give an endurance ratio of 20000/96800 or 0.21. The ratio is below that found for the propulsion shaft steel reported previously, and is quite low for any type of steel.

These test results, as well as others not reported herein, indicate that severe longitudinal inclusions do not lower the rotating cantilever flexural fatigue properties of steel. The alternating torsion properties of half inch specimens appear to be lowered 20% to 25% at the most. The inclusions in such cases are distributed at random in the steel.

Longitudinal Inclusions With Preferred Distribution. The second type of inclusions to be considered are also longitudinal, but

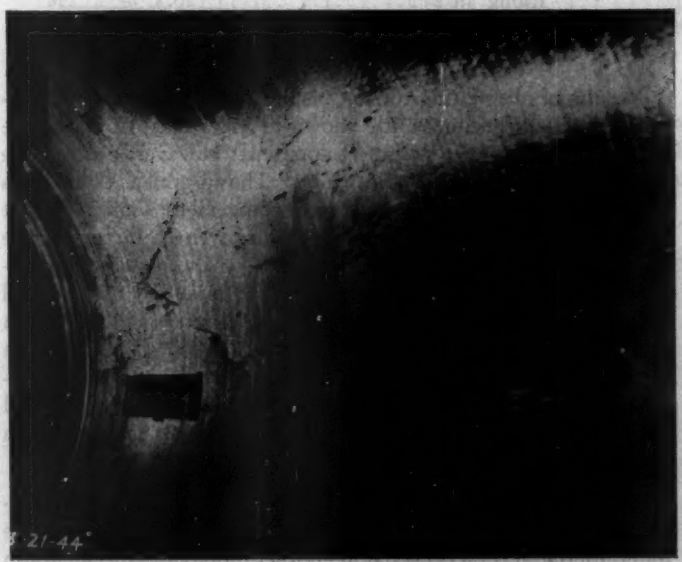


FIGURE 8.

occur in a preferred distribution other than longitudinal. An example of this type was found in the examination of two sections of 16 inch propulsion shafting. Figure 8 is a photograph of the bore of one of the shafts. It is at once apparent that a large number of longitudinal inclusions are present in a staggered arrangement running about 45 degrees from the longitudinal. The longest inclusion found was 0.6 inch in length, whereas the average length was 0.25 inch.

The average composition and static properties of the shaft, which is a molybdenum-vanadium steel, were found to be as follows:

C	0.35 %	Si	0.20%
S	0.028	Mo	0.45
P	0.012	V	0.18
Mn	0.78		
Tensile Strength	83,800	psi	
Yield Strength, .1% Set	56,800	psi	
Proof Stress	56,300	psi	
Elong. in 2"	23.2	%	
Reduction of Area	39.2	%	
Brinell Hardness	163		

Results of alternating torsion tests, made with inclusion bearing 0.500 inch specimens, are shown in Figure 9. Considerable scattering of the test points is apparent; however, an average stress-cycle curve has been drawn in. The specimens themselves contained inclusions in a staggered arrangement similar to that noted in the original forging. This apparently increased the reduction to the fatigue strength, because the distribution followed the approximate 45 degree pattern taken by any normal torsional fatigue fracture. The endurance limit for the inclusion bearing specimens is estimated to be 15,500 psi. This gives an endurance ratio of 15500/83800 or 0.185. The ratio is extremely low and demonstrates that several inclusions arranged along a single stress plane are more damaging than a single inclusion.

Globular Inclusions. The third type of inclusions to be considered can be classified as globular. Perhaps the best example found by the Station occurred in another molybdenum-vanadium steel propulsion shaft. Figure 10 is an etched cross section show-

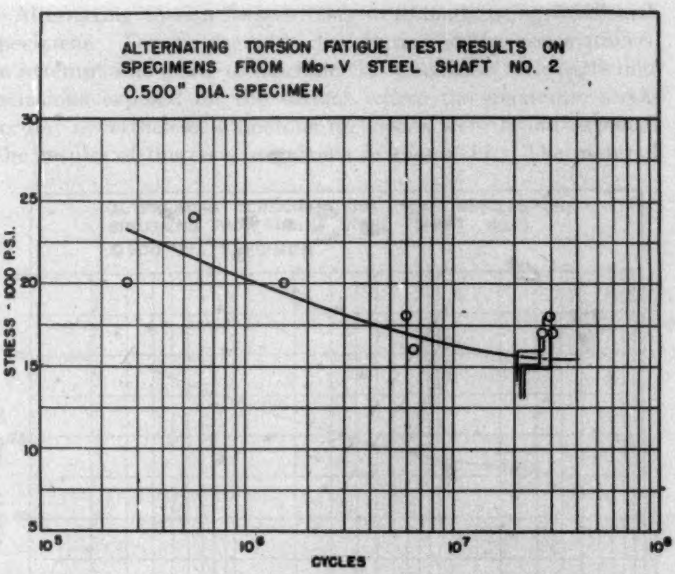


FIGURE 9.

ing the general distribution of inclusions. Metallographic examinations revealed structures like that shown by Figure 11. The inclusions tended to occur in patches, and were globular whether viewed in the longitudinal or transverse directions.

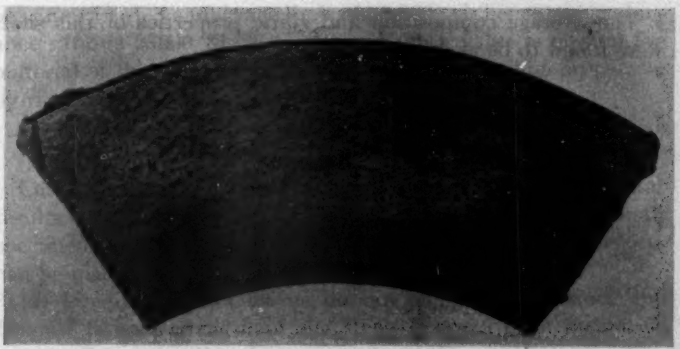


FIGURE 10.

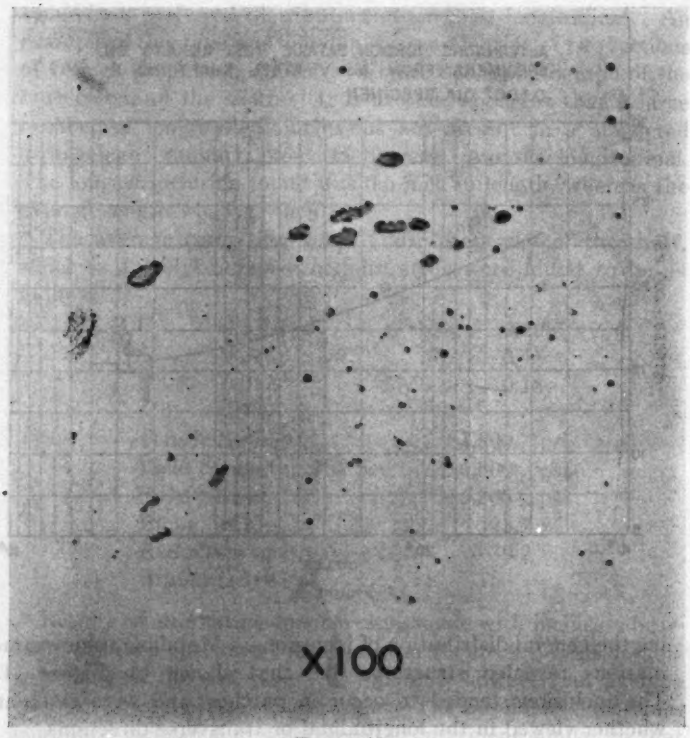


FIGURE 11.

The average composition and static properties of this shaft were found to be as follows:

C	0.44 %	Cr	None
S	0.033	Ni	0.01%
P	0.05	V	0.30
Mn	0.63	Mo	0.44
Si	0.24		

Tensile Strength	81,300	psi
Yield Strength, .1% Set	50,400	psi
Proof Stress	49,000	psi
Elongation in 2"	25.5	%
Reduction of Area	52.1	%
Brinell Hardness	222	

Alternating torsion fatigue tests were made using 0.500 inch specimens. Due to the wide distribution of the non-metallics, no attempt was made to machine the specimens with particular inclusions exposed on the surface where the maximum stress occurs; nevertheless, numerous inclusions were found exposed. The results of the tests are shown in Figure 12. The material

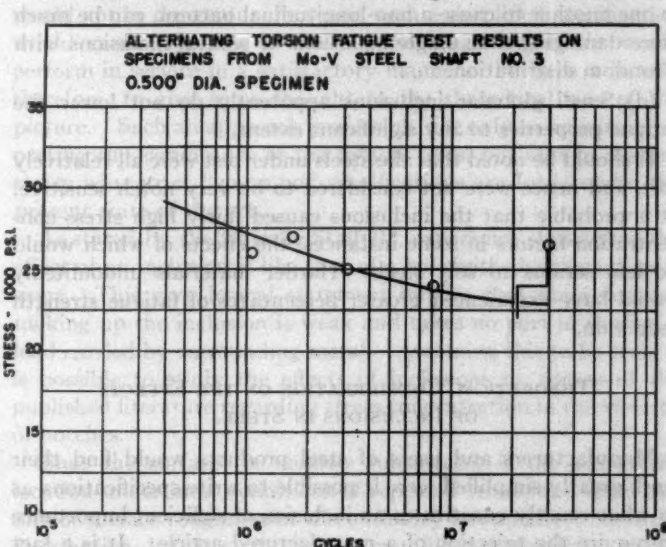


FIGURE 12.

has an estimated endurance limit of 23,000 psi; hence, the endurance ratio is 23000/81300 or 0.28. Such a value is normal for material of this type, and any lowering of the fatigue properties by the inclusions was very slight if at all.

CONCLUSIONS DRAWN FROM EXPERIMENTAL RESULTS.

Certain conclusions can be drawn from the experimental results which have just been presented. For tests made on small specimens, mostly ½ inch in diameter, the following statements can be made:

(a) Longitudinal inclusions do not significantly reduce the flexural fatigue strength of the steels tested. Such inclusions are parallel to the direction of stress.

(b) Large longitudinal inclusions apparently can reduce the torsional fatigue strength of steel. The inclusions which were studied were sufficiently large and numerous to cause hesitancy in accepting the forgings for service. Such inclusions lowered the torsional fatigue strength 20 to 25 per cent.

(c) A number of longitudinal inclusions, arranged with respect to one another to cause a non-longitudinal pattern, can be much more damaging than single inclusions or several inclusions with a random distribution.

(d) Small globular inclusions apparently do not lower the fatigue properties to any significant extent.

It should be noted that the steels under test were all relatively soft, and hence were not considered to be very notch sensitive. It is probable that the inclusions caused fairly high stress concentration factors in some instances, the effects of which would be less serious to soft steels. Harder materials undoubtedly would have experienced greater percentages of fatigue strength reduction.

Theoretical Consideration on the Effects of Inclusions in Steel.

Manufacturers and users of steel products would find their work greatly simplified were it possible to write specifications as to what exactly constitutes an inclusion of sufficient importance to require the rejection of a manufactured article. It is a fact that inclusions must be considered as possible sources of danger, because numerous fatigue failures have been reported in the literature as directly associated with abnormal inclusion conditions. Despite all the experimental work, however, fatigue tests have been able to supply only very generalized conclusions at the best. The Station's tests are no exceptions, and it is quite apparent that the tests described above are not capable of yielding enough information for writing an all-inclusive specification.

It is common knowledge that all steels contain inclusions. These may vary in size from submicroscopic particles to large non-metallics like those which have been described. It is probably true that the extent to which an inclusion can cause damage depends on its size, other factors, such as material, location,

orientation, etc., being equal. Unfortunately, these other conditions never are equal in actual practice. As a result, there is very little hope of ever writing specifications which can rigorously defend a limitation on the size of an acceptable inclusion. It might be possible to be very conservative by establishing limits below which an inclusion, under the most adverse conditions, would not be dangerous; however, such a simplified specification would cause rejection of many parts which ordinarily would perform in service in a satisfactory manner. Perhaps it is well, therefore, to give some thought to the theoretical side of the picture. Such an approach is of value, not only as help in interpreting the significance of the fatigue tests, but also to demonstrate why strict "go-no go" specifications are futile under the present state of the art.

As stated by the Battelle Staff(1)*, experience shows that the effect of an inclusion is like that of a hole with the same dimensions. This very logical statement implies that the substance making up the inclusion is weak and takes no part in sharing a load carried by surrounding metal. Assuming this to be true, it is possible to study the effects of inclusions by means of the published literature regarding stress concentration in the vicinity of notches.

Since the surface (or shallow subsurface) is the vulnerable location under most conditions of repeated stress, and since any sort of surface roughness or discontinuity has an effect on stress distribution, it follows that a non-metallic inclusion lying at the surface may be deleterious. As in all other instances, such discontinuities are more serious in harder steels, because the *notch sensitivity* varies according to the material. On the other hand, the *stress concentration factor* theoretically is not a function of the steel; it is a function only of the size, shape and relative location of the notch. For the time being, the discussion will be limited to the stress concentration factor alone. Further remarks regarding sensitivity will be reserved till later.

A large amount of published information is available on stress concentration factors introduced by a variety of notches. These have been summarized adequately through 1942 by Roark(2) in his recent book on stress and strain. Nevertheless, there are two very important reasons why these formulas cannot be used in

*Figures in parentheses refer to the references appended to this paper.

their present forms for the study of inclusions. The two reasons are best explained with an example. The formula quoted by Roark from the data of W. K. Wilson will suffice; this formula gives the stress concentration factor introduced by a keyway in a circular shaft under torsional stress, as follows:

$$K' = 1 + \sqrt{\frac{t}{r}}$$

where t = depth of the keyway
and r = radius in the corner at the bottom of the keyway.

It is apparent that a longitudinal inclusion exposed on the surface of a shaft resembles closely the conditions described by the formula. Incidentally, before going further, it is well to state that this formula is valid only for the case of "shallow notches", that is, where only the shape and depth of the notch are controlling factors, and in which the diameter of the shaft is large and unimportant with respect to the concentration factor obtained. It is apparent that inclusions in large forgings fulfill these requirements. An examination of the formula discloses that factor K′ increases as the radius decreases (i.e., as the notch becomes sharper) and approaches infinity as the radius approaches zero. Obviously it is not possible for the stress concentration factor to reach astronomical values just because the notch becomes very sharp. This inherent weakness applies to nearly every formula for stress concentration factors up to the time of Neuber[8], and thereby limits the use of such equations in their present form to notches with relatively large radii of curvature. More will be said later on Neuber's contribution in this connection.

Continuing the discussion of the formula, it should be mentioned that nearly all equations for stress concentration contain the ratio t/r as a controlling term. This is important in dealing with non-metallics as they occur in steel forgings. It is apparent that the radius r decreases as the inclusion size becomes smaller. This does not mean necessarily that the stress concentration increases, because the depth of the inclusion t also decreases at the same time as the radius. If an inclusion occurs at the surface where the major damage is to be expected, the value of t can never exceed the dimension of the inclusion measured along the line leading from the neutral axis of the machinery component.

In fact, for all inclusions shaped similarly except for size, the value of the ratio t/r is a constant. It is interesting to observe that if the transverse cross section of an inclusion is circular, and the inclusion is barely exposed on the surface of the part, the maximum value which can be attained by the ratio t/r is 2. Such an inclusion, if applied in the above formula for a shaft under torsional stress, would yield

$$K' = 1 + \sqrt{2} = 2.4,$$

and, without Neuber's correction, would indicate the same stress concentrating effect regardless of the size of the non-metallic.

Now it is known from experience that the above is not the case and that large inclusions definitely cause greater fatigue damage than do small inclusions. In fact, all steels contain inclusions, and those containing normal amounts ordinarily are not considered to be damaged at all. The explanation for this can be found in the treatment by Neuber([3]), whose work remained unpublished in this country until 1945. A brief explanation of this work will be adequate.

Whereas the classic theory of elasticity suffices for notches whose radius of curvature is not too small, Neuber recognized that it yields entirely incorrect results for pointed notches or even for strongly curved notches. Thus, for example, the value

$$1 + \sqrt{\frac{t}{r}}$$

was found for the stress concentration factor of shallow notches under shear or torsion (note that Neuber derived this value theoretically, and it corresponds to the empirical value quoted by Roark). For instance, if the radius of curvature were only 1/10,000 of the depth, the stress concentration factor would have the impossible value of 101. Conditions are similar for all the stress concentration factors derived. For a radius of curvature which is vanishingly small, the concentration factor becomes proportional to

$$\sqrt{\frac{t}{r}}$$

in all cases. Accordingly, it would assume the impossible value of infinity for pointed notches. Obviously, the limit of the range of validity of traditional theory has been reached. On the other

hand, so long as the problem deals with weakly curved notches, in which the curvature is everywhere large enough compared to the crystalline structure of the material, no error becomes evident, and the usual formulas can be applied satisfactorily.

In consideration of these facts, Neuber introduced a new concept, the reasoning for which need not be presented here. It is sufficient to state that a formula was developed which accounts quite well for experimental observations in the transition from weakly curved to pointed notches. This formula is recorded conveniently as follows:

$$K = 1 + \frac{K'-1}{1 + \frac{\pi}{\pi - W}\sqrt{\frac{r'}{r}}}$$

where K = technical stress concentration factor.

K′ = stress concentration factor of an ideal material (obeys classic theory of elasticity). (Note that K′, therefore, has been applied properly in the formula quoted by Roark.)

W = flank angle of the notch.

r = radius of curvature of the notch.

r′ = elastic notch sensitivity (related to the texture of the material) and reported to equal 0.0189″ for steel.

The flank angle of the notch is not a constant when dealing with inclusions, because the non-metallics would occur with random distribution and would be exposed to varying extents on a machined surface. Note, however, that K is greatest when the flank angle is zero. For all practical purposes, therefore, when dealing with rounded inclusions, the greatest damage would occur if half or more of the inclusion were left in the surface after machining. Consequently, for the purpose of this discussion, it is well to assume the worst possible case by letting the flank angle W be zero. Further, since the discussion is concerned primarily with steels, the term r′ can be accepted as 0.0189 inches. The above formula then reduces to

$$K = 1 + \frac{K'-1}{1 + \sqrt{\frac{.0189}{r}}}$$

The precision and the validity of such an equation under rigorous examination are open to question and subject to further experimental confirmation. Nevertheless, the general theory is sound and will serve very well to explain why small inclusions are less damaging than large ones. The trend can be illustrated adequately by plotting a curve, once again using the equation for a longitudinal notch with a circular cross section occurring at the surface of a solid circular shaft under torsional stress. The equation is

$$K' = 1 + \sqrt{\frac{t}{r}}$$

followed by the correction introduced by Neuber. In this example, the case shall be treated in which the size of the inclusion varies and in which half of the inclusion is machined away during the finishing of the shaft. The size of the inclusion shall be small in relation to the size of the shaft. Typical calculations using the above equations are presented in Table II.

TABLE II.
CALCULATIONS FOR THE TECHNICAL STRESS CONCENTRATION FACTOR FOR A LONGITUDINAL INCLUSION WITH A CIRCULAR CROSS SECTION, HALF EXPOSED ON THE SURFACE OF A SOLID CIRCULAR SHAFT UNDER TORSION

Radius of Cross Section of Inclusion Inches	Depth of Inclusion Inches	Stress Concentration Factor Calculated as an Ideal Material	Technical Stress Concentration Factor Using Neuber's Equation
r	t	K'	K
.25	.25	2.00	1.785
.15	.15	2.00	1.739
.10	.10	2.00	1.696
.05	.05	2.00	1.620
.03	.03	2.00	1.557
.01	.01	2.00	1.421
.005	.005	2.00	1.340
.002	.002	2.00	1.245
.001	.001	2.00	1.187
.0005	.0005	2.00	1.140
Zero	Zero	1.00	1.000

Graphical presentation of the data in Table II is presented in Figure 13. Due consideration should be given to the change occurring in the abscissa scale. It is of interest to note that the

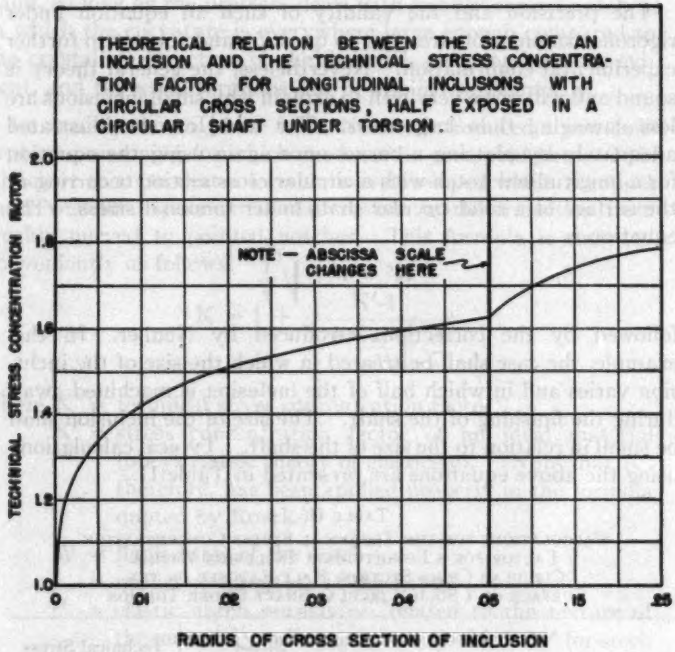


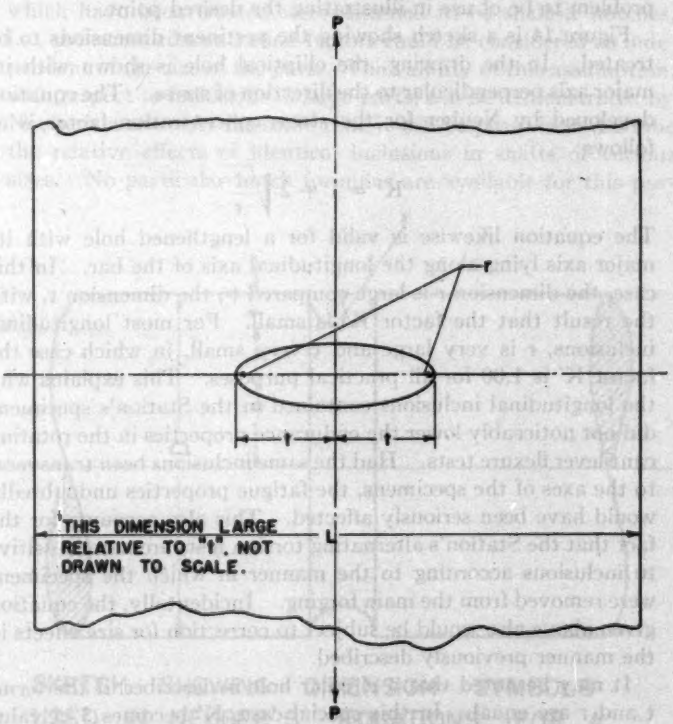
FIGURE 13.

theory predicts relatively small change in the stress concentration factor over quite a range of large sized inclusions. This is evidenced by the fairly horizontal position taken by the curve as the inclusion size becomes large. On the other hand, as the inclusions become very small, the stress concentration drops sharply, becoming 1.000 (no concentration) as the size of the inclusion becomes vanishingly small. It is very likely that the critical sizes shown on the curve are in considerable error; however, the curve has been presented only to illustrate the general trend to be expected.

The foregoing discussion has thrown considerable light on the question regarding the effects of the size of an inclusion. The trend shown by the example above would be similar in other cases in which the members might be stressed in different ways and in which different shapes of inclusions might be present.

Hence, the example, besides being specific, represents the general case as well.

The discussion now will be shifted to a consideration of the effects of inclusion orientation with respect to the direction of stress. As before, the case to be treated will be that of longitudinal inclusions. Again it shall be assumed that the inclusions are small in relation to the size of the part, so that the problem becomes one concerned only with shallow notches, and not one concerned with the overall dimensions of the part; this assumption is entirely valid for inclusions in large forgings. When these



ELLIPTICAL HOLE IN A VERY WIDE RECTANGULAR SPECIMEN UNDER TENSION

FIGURE 14.

conditions exist, the stress concentration factor, introduced by a small notch on the outside of a large shaft, will be the same whether the shaft be under axial tension or axial bending stresses(3).

No specific formulas have been presented in the literature to deal with lengthened surface discontinuities on the outside of circular members under tensile loads. However, Neuber presents a complete development for an elliptical hole occurring in a very wide rectangular bar (shallow notch condition) under tension. This development is allied closely enough to the present problem to be of use in illustrating the desired point.

Figure 14 is a sketch showing the pertinent dimensions to be treated. In the drawing, the elliptical hole is shown with its major axis perpendicular to the direction of stress. The equation developed by Neuber for the stress concentration factor is as follows:

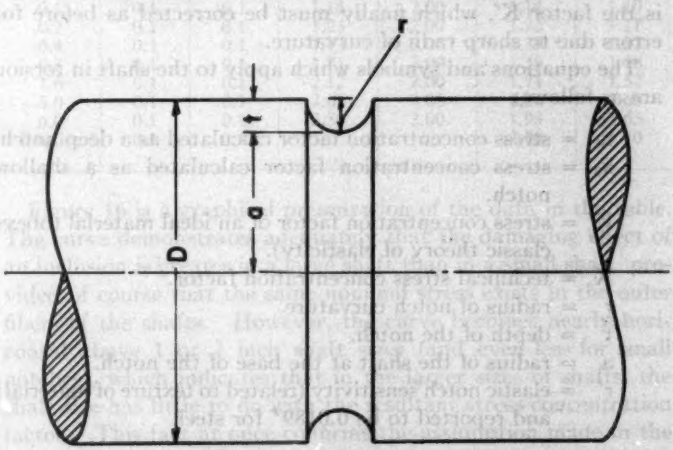
$$K' = 1 + 2\sqrt{\frac{t}{r}}$$

The equation likewise is valid for a lengthened hole with its major axis lying along the longitudinal axis of the bar. In this case, the dimension r is large compared to the dimension t, with the result that the factor K′ is small. For most longitudinal inclusions, r is very large and t very small, in which case the factor K′ is 1.00 for all practical purposes. This explains why the longitudinal inclusions contained in the Station's specimens did not noticeably lower the endurance properties in the rotating cantilever flexure tests. Had the same inclusions been transverse to the axes of the specimens, the fatigue properties undoubtedly would have been seriously affected. This also accounts for the fact that the Station's alternating torsion tests are more sensitive to inclusions according to the manner in which the specimens were removed from the main forging. Incidentally, the equation given above also would be subject to correction for size effects in the manner previously described.

It may be noted that a circular hole is described if the terms t and r are equal. In this special case, K′ becomes 3, a value which in turn is subject to correction for small hole sizes according to the method described above. The factor of 3 probably is not correct when applied to spherical inclusions, since the inclusions are not accurately treated as holes in flat plates. Nevertheless,

the similarity is good enough to conclude that spherical inclusions cause some degree of stress concentration. Apparently the effect is not great, and in the case of the specimens with globular inclusions which were tested at the Station, the size was small enough to reduce the stress concentration factor to a very insignificant value.

The discussion so far has demonstrated the importance of inclusions from the standpoint of size and orientation with respect to the direction of stress. One further point now will be considered. It has been mentioned previously that the notches which have been treated were assumed to be shallow notches, i.e., the stress concentration factors could be considered as independent of the size of the part. The validity of this assumption, with respect to inclusions in large parts, can be demonstrated by Neuber's work. At the same time it will be possible to describe the relative effects of identical inclusions in shafts of various sizes. No particular notch formulas are available for this pur-



SKETCH SHOWING DIMENSION SYMBOLS USED IN NEUBER'S EQUATIONS FOR A SOLID CIRCULAR SHAFT IN TORSION. (FLANK ANGLE OF NOTCH ZERO DEGREES)

FIGURE 15.

pose which might resemble the notches caused by inclusions. However, this is no handicap to the purpose of the discussion because all types of notches occurring in shafts obey the same general pattern. Therefore, to illustrate the point in question, the equations dealing with a circumferential notch in a solid circular shaft under torsion will suffice.

One of the chief advantages to the work of Neuber rests in the fact that stress concentrations can be calculated for notches of any radius of curvature and of any depth, occurring in any size of shaft. Figure 15 shows the dimensions which apply in the case of a shaft under torsional stresses. Briefly, the method consists of calculating the stress concentration factor considering the notch as a shallow notch (involving only r and t) and the stress concentration factor considering the notch as a deep notch (involving only a and t). These two factors in turn are used to calculate the stress concentration factor for a notch of arbitrary depth, thereby involving all three terms, r, t and a. This value is the factor K′, which finally must be corrected as before for errors due to sharp radii of curvature.

The equations and symbols which apply to the shaft in torsion are as follows:

k_1 = stress concentration factor calculated as a deep notch.
k_2 = stress concentration factor calculated as a shallow notch.
K′ = stress concentration factor of an ideal material (obeys classic theory of elasticity).
K = technical stress concentration factor.
r = radius of notch curvature.
t = depth of the notch.
a = radius of the shaft at the base of the notch.
r′ = elastic notch sensitivity (related to texture of material) and reported to be 0.0189″ for steel.

$$k_1 = \frac{3\left(1+\sqrt{\frac{a}{r}+1}\right)^2}{4\left(1+2\sqrt{\frac{a}{r}+1}\right)}$$

$$k_2 = 1+\sqrt{\frac{t}{r}}$$

$$K' = \frac{(k_1-1)\,(k_2-1)}{\sqrt{(k_1-1)^2 + (k_2-1)^2}}$$

$$K = 1 + \frac{K'-1}{1 + \sqrt{\frac{r'}{r}}}$$

Typical calculations using these equations are presented in Table III. The notch chosen for study contains both the radius and depth equal to 0.1 inch.

TABLE III.
CALCULATIONS FOR TECHNICAL STRESS CONCENTRATION FACTORS FOR A CIRCUMFERENTIAL NOTCH IN A SOLID CIRCULAR SHAFT UNDER TORSION.

a	t	r	k_1	k_2	K′	K
0	0.1	0.1	1.00	2.00	1.00	1.00
0.1	0.1	0.1	1.14	2.00	1.14	1.10
0.2	0.1	0.1	1.25	2.00	1.25	1.17
0.4	0.1	0.1	1.43	2.00	1.39	1.27
0.8	0.1	0.1	1.71	2.00	1.58	1.40
1.6	0.1	0.1	2.12	2.00	1.74	1.52
3.0	0.1	0.1	2.67	2.00	1.86	1.60
6.0	0.1	0.1	3.50	2.00	1.93	1.65
infinite	0.1	0.1	....	....	2.00	1.70

Figure 16 is a graphical presentation of the data in the table. The curve demonstrates adequately that the damaging effect of an inclusion is greater in a large shaft than in a small shaft, provided of course that the same nominal stress exists in the outer fibers of the shafts. However, the curve becomes nearly horizontal above 1 or 2 inch shaft sizes (and even less for small notches), which indicates that in the larger sizes of shafts, the shaft size has little to do with the resultant stress concentration factor. This fact at once confirms the assumption made in the earlier treatment of notches, that is, that in dealing with inclusions in large forgings it is necessary only to treat the notches as shallow defects which are independent of the size of the part.

GENERAL CONCLUSIONS.

The experimental results and the discussion of the theory have led to several general conclusions regarding the effects of inclu-

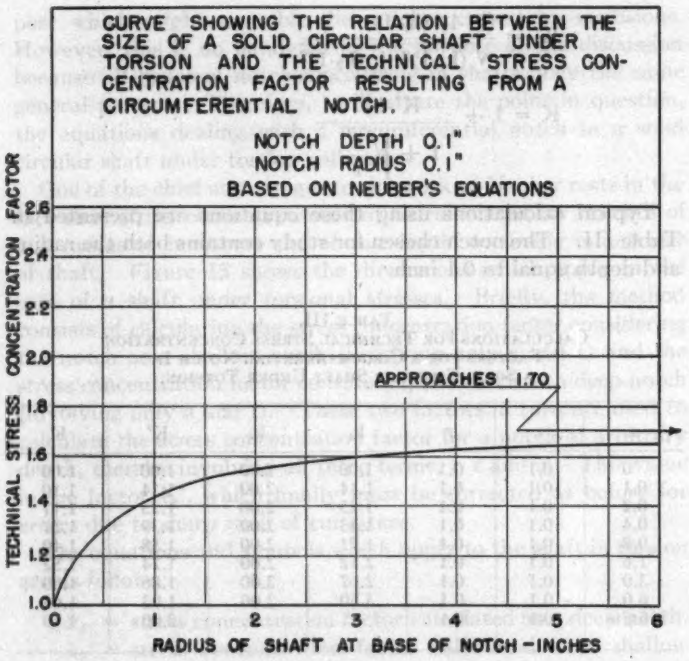


FIGURE 16.

sions on the endurance properties of steel, particularly in large forged parts. These conclusions are presented as follows.

The inclusions which are most serious in machinery components usually will be those exposed on the surface where stresses are highest and where fatigue failures generally originate. Fortunately this is the location where inspection procedures are the easiest to perform.

Severe inclusions in the surface of a steel part can lower the fatigue resistance considerably. However, if the maximum stress on the surface of the part is not uniform in itself, then the location of the inclusion will determine to a large extent whether or not the part is serviceable. For example, if an inclusion occurs in a fillet, keyway, oil hole or other change of section, it would be more undesirable than for one occurring where stress concentrating conditions do not already exist.

A knowledge of the stress conditions which exist in a part at the region of an inclusion would be very helpful in deciding the serviceability of a machinery part. This is particularly true for large forgings such as propulsion shafts. In such shafts the maximum service stresses are relatively low and probably would be below the endurance limit of the steel despite the inclusions. It might be mentioned that not a single case has ever come to the attention of the Station in which inclusions caused the failure of a large propulsion shaft. This is not intended to imply that all shafts with large inclusions should be accepted, and very possibly some of the shafts which have been rejected would have failed had they been used in service.

Large forgings usually are made of relatively soft steels. Such materials are expected to be rather insensitive to notch conditions. Hence, inclusions are more serious if they occur in hard steels.

The Battelle Staff(1) has made the statement that "wherever the difference between a mirror-lapped surface and a rough ground surface would affect the service life under repeated stress sufficiently to demand polishing the surface, then avoidance of even ordinary small inclusions and the selection of extraordinarily clean aircraft quality steel is fully justified." This statement supports the fact that the intended service and stresses destined for a machinery part control to a large extent the necessary cleanliness of the steel.

The Orientation of an inclusion, with respect to the direction of stress, influences to a marked degree the resulting stress concentration. Long narrow inclusions lying in the direction of stress have practically no effect. The same inclusions lying perpendicular to the direction of stress may exert a marked influence. This accounts for the fact that lengthened inclusions running parallel to the axis of a shaft have no significant effect on the flexural properties, but do have an effect on the torsional properties.

Globular inclusions cannot be said to have any special orientation with respect to the direction of stress. Such inclusions are the least serious kind, and if not very large can be considered as having little effect on the endurance properties of unhardened steels.

The present theory confirms experience by demonstrating that small inclusions are less damaging than large inclusions, other things being equal. Very little difference in stress concentrating effects occur over a broad range of large inclusion sizes. However, as the inclusion size gets small, there is a marked decrease in stress concentration. The present state of the art does not indicate exactly what the critical size is. About all that can be said is that normal sizes of inclusions, as they occur in the greatest percentage of steel products, are small enough to cause no significant effects on the endurance properties.

The stress concentration caused by an inclusion in the surface of a shaft increases as the shaft size increases. However, the increase is very slight over a wide range of large shaft sizes. Conversely, as the shaft size gets relatively small, there is a rapid decrease in the stress concentration. This, of course, indicates that an inclusion in a large shaft cannot be ignored merely because the size of the inclusion appears offhand to be insignificant relative to the size of the shaft. The important consideration is the nominal stress to be applied to the shaft which may be further increased locally by the non-metallic defect.

The state of the art at present does not allow for rigid specification requirements to be written on the acceptable limits of non-metallics in large forged parts. It appears necessary to recommend that doubtful cases must be considered individually by those qualified to make sound decisions on the serviceability of an inclusion bearing part.

This report has been concerned only with non-metallic inclusions. Flakes, as they appear in large steel parts, are known to be far more serious than inclusions and should be considered adequate basis for the rejection of a forging. Care must be exercised in differentiating between flakes and inclusions during inspection.

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DYNAMIC STRENGTH.

By Manley St. Denis.

"Time is a necessary dimension of strength."

1. Preface.

The external loadings on a vessel, arising either from the interplay of the elements or thru the action of man, are dynamic in nature. Yet in all strength calculations this dynamic aspect is treated superficially, if at all. The result is that in many instances the correlation is lacking between calculations and experience. In these instances proportioning of scantlings, prediction of strength and explanation of failures are based at best on empirical rules. These are generally the result of costly experiments or experience and cannot be relied upon except under conditions of similarity which unfortunately seldom if ever exist. Until a method is made available for designing structures subject to dynamic loading, proper design will be an elusive undertaking and will largely depend on arbitrary individual opinion. But to have a method of design is not sufficient. To be of utmost value the method must be simple so that it can readily be applied in the drafting room. In this paper is presented a first simple approximation to a theory of dynamic

The Author.

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After working one year in the field of Civil Engineering and three years in the Hull Scientific section of the New York Shipbuilding Corporation, he went to the Pearl Harbor Naval Shipyard. Three years later he was transferred to the Bureau of Ships where he remained for nine years, during the last six of which he was head of the Hull Scientific Section. In April of this year he was transferred to the Hydromechanics Department of the David Taylor Model Basin and is now in charge of Performance Predictions.

strength applicable to the design of ship's structures. The examples include evaluation of dynamic load factors for:

(*a*) Flight deck structure with aircraft landing.

(*b*) Deck structure subjected to gun blast.

2. INTRODUCTION.

It has long been known that it is not the severest sea that subjects a vessel to the severest strains. Yet explanations advanced for this apparent paradox have not been such as to permit a quantitative evaluation of the phenomenon. In ordinary calculations the primary strains in the ship's girder are assessed by integration of the weight curve and wave profiles. For this purpose the vessel is assumed poised in static equilibrium on a stationary standard wave equal to its length. From a purely static consideration of forces and moments this assumption is, or approaches, the critical. A ship designed to develop reasonable strains when acted upon by the sea in such a manner should not experience structural distress when in a seaway of smaller force. Indeed, if primary strains are calculated for the sea condition at which a structural distress was experienced, it will generally be found that the result is inconsistent with the behavior of the structure.

In the case of the ship's girder just mentioned structural failures usually occur at apparently low calculated stresses. In contrast to this there exist other cases in which structural failures do not occur in spite of apparently high calculated stresses. An example of the latter is the resistance of decks and superstructure to blast pressure from ship's own guns. The peak blast pressure to which structure in the vicinity of the muzzle is subjected may be of a magnitude several times that at which collapse should occur. Yet records of structural firing trials rarely reveal any damage, and then only to secondary structure.

The list of cases for which this lack of relationship apparently exists between the external loading on a structure and its behavior can be extended to include projectile impact, flight deck landings, reaction of turret structure to gun recoil, under-water explosions, etc. All these cases have in common one distinguishing feature, namely, that the external loading is dynamic, that is, the externally applied load varies rapidly with time. One can no

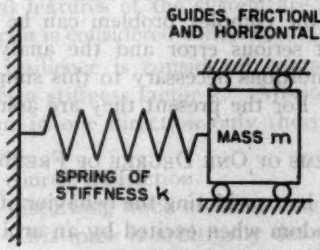
longer consider the loading as a simple steady force of a given magnitude but rather as a pulse characterized by growth, duration and decay.

The question naturally arises as to how the introduction of the time element affects the behavior of the structure. We will find that under dynamic loading the behavior of a structure is no longer defined solely by considerations of external loading and of elasticity as under static conditions, but that it becomes necessary to introduce the element of inertia.

The purpose of the present paper is to examine the relation between an external dynamic loading on a structure and the resulting strains therein, and to provide formulas for predicting with a fair degree of accuracy its elastic behavior.

3. Definition of the Problem.

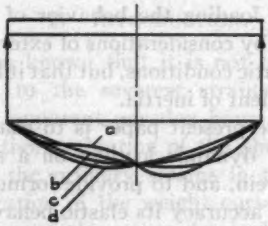
The problem is, at the outset, most complex, for the structure in which we are interested, such as a ship or component parts thereof, do not constitute simple elastic systems which can be analyzed without excessive effort. Yet, the behavior of such structures can best be understood thru a study of simple elastic systems. It is, therefore, wise to begin by reducing the problem to its simplest terms and considering, at first, systems having a single degree of freedom, that is, systems whose position at any instant can be completely described by a single parameter. The classical illustrative problem is that of a weight acted upon by a spring and constrained by guides to move in a vertical or horizontal direction (Figure 1).



SYSTEM HAVING ONE
DEGREE OF FREEDOM

FIGURE 1

A flexible beam possesses theoretically an unlimited number of degrees of freedom. By suitable external loadings (providing the strength of the beam is not exceeded) it is possible to deflect a beam into a curve on any shape (Figure 2), for the complete



POSSIBLE MODES OF VIBRATION OF A BEAM

FIGURE 2

definition of which an unlimited number of parameters is required (even in the two-dimensional case). The same reasoning applies with even greater strength to plates.

It is self evident that a structure whose component parts possess unlimited degrees of freedom can of itself have no less. This is the case with all ship's structures.

However, it frequently occurs that in a structure freedom in a single direction is preponderant to such an extent that for practical purposes it determines the behavior of the system. When this occurs, the complex problem can be reduced to a simple one without serious error and the analysis is greatly simplified. The conditions necessary to this simplification will be discussed later. For the present they are assumed to exist.

4. Systems of One Degree of Freedom.

Suppose we begin by considering the behavior of a system of a single degree of freedom when excited by an arbitrary external force variable with time.

The assumptions are made that:

(*a*) The system is linear, i.e., the deformations obey Hooke's law.

(*b*) The exciting force is not severe enough to strain the system beyond the elastic limit.

(*c*) Internal damping is negligible.

The first two assumptions are common to all problems in theory of elasticity.

The last assumption needs be discussed somewhat more fully. It is, of course, well known that when a simple elastic system is acted upon by a periodic force, a condition of resonance results when the period of the exciting force approaches or coincides with the natural period of vibration of the system. If there is no internal damping the amplitudes resulting during resonance tend to infinity. But introduction of damping (even in a small amount) imposes an upper boundary to the amplitudes of the system. When the system, therefore, is excited by a periodic force, its behavior during resonance is essentially determined by the amount of internal damping. When the excitation is no longer periodic the behavior of the system follows a more intricate pattern. Resonance still can occur, altho the prediction is no longer so simple. But damping ceases to be significant in view of the fact that the amplitudes of the system cannot build up when the period of the exciting force is continually varying.

We are particularly interested in temporary excitations lasting no longer than a single cycle of the natural period of vibration. Such temporary excitations are called transients. It is self-evident that internal damping in the case of transients does not affect the problem.

The essential features of the system are shown in Figure 1. The whole inertia is considered concentrated in the mass m.

The whole resilience is considered contained in the spring which possesses a stiffness factor k. Motion is constrained by guides to occur in one direction only (horizontally) along the axis of abscissae.

There is, of course, no friction.

An external disturbing force of finite magnitude P is applied to the mass. This force is arbitrarily defined and varies with time.

The problem is to obtain the resulting reaction R in the spring as a function of time. Because the system is linear, the reactive force for any displacement x is equal to kx. Accordingly, it is

sufficient to determine only the motion of the system as a function of time.

The equation of motion is simply derived. Application of the external force, P, gives rise to a resilience kx and an inertia force $m\frac{d^2x}{dt^2}$ both acting opposite to P. For equilibrium

$$m\frac{d^2x}{dt^2} + kx = P \qquad (1)$$

Both x and P are functions of t only and may be written x(t) and P(t) respectively.

In order to bring out more forcibly the difference in behavior of the system under static and dynamic loadings, it will be convenient to express the results in non-dimensional form.

To this effect, in lieu of the disturbing force P we introduce its ratio at any instant to the peak load P_o. This first ratio $\frac{P}{P_o} = F$ or F(t) is termed the disturbance.

Similarly instead of the displacement x we introduce its ratio at any instant to the displacement x_o under the steady load P_o This second ratio, $\frac{x}{x_o} = X$ or X(t), is termed the response. Obviously, for the steady condition, the response is unity. With these substitutions by dividing (1) thru by $kx_o = P_o$ the equation can be rewritten in non-dimensional form, as

$$\frac{1}{a^2}\cdot\frac{d^2X}{dt^2} + X = F(t) \qquad (2)$$

where

$$a^2 = \frac{k}{m}$$

This is the fundamental equation defining the motion of an undamped linear system of a single degree of freedom acted upon by an arbitrary disturbance.

5. Solution of the Differential Equation. General Case.

This differential equation can be solved in a variety of ways, some of which are particularly elegant. In this paper use will be made of the Laplace-transformation method because of its

extreme simplicity when applied to problems of the nature being discussed (transients in mechanical systems). The advantage of the Laplace-transformation method over the classical method or solving differential equations lies in the ease with which a solution is reached for a particular set of initial conditions. In contrast thereto, the classical method, which requires that a general solution be first obtained and then specialized to fit a particular set of conditions, is slow and clumsy.

The differential equation is solved in the Appendix for certain forms of disturbance. It may be noted here that the complete solution of the differential equation can always be considered as consisting of two partial solutions, sometimes termed respectively the complementary function and the particular integral.

The complementary function gives us the response when there is no disturbance, i.e. after the application of a pulse. As shown in the Appendix, this solution is obtained by letting the forcing term $F(t) = 0$ in equation (2) and may be written as:

$$X = X_o \cos at + \frac{1}{a}\left(\frac{dx}{dt}\right)_0 \sin at \qquad (3)$$

where the subscript 0 applies to the initial conditions for $t = 0$.

If the initial conditions are taken at any other time, $t = t_1$ for example, the equation assumes the form:

$$X = X_1 \cos a(t - t_1) + \frac{1}{a}\left(\frac{dx}{dt}\right)_1 \sin a(t-t_1) \qquad (4)$$

This latter equation, consequently, defines the displacement of the system at any time t, providing the displacement and velocity of the system are known at a previous time t_1, and providing further that no forcing term operates in the interval t_o to t.

The presence of a disturbance gives rise to additional solutions (the particular integral) which in the Appendix are shown to be expressible as:

$$X = a\int_o^t F(t) \sin a(t-\tau)\, d\tau \qquad (5)$$

where τ is the variable of integration between the limits of 0 and t. In lieu of integrating this expression, however, it is more convenient to solve the differential equation directly for the desired disturbance as carried out in the Appendix.

It should be noted that for a system initially at rest, the motion in the interval during which a pulse operates is defined by the particular integral alone. Subsequently to the pulse, the motion of the system is defined solely by the complementary function, but care must be taken to insure that the initial conditions applying to the complementary function are those existing at the instant of removal of the pulse.

We may now consider specific forms of disturbance, and, since we are interested in transients, we will consider only disturbances that are momentarily applied, to wit:

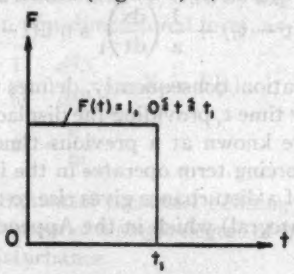
(*a*) The step pulse
(*b*) The sine pulse
(*c*) The blast pulse.

In the Appendix further cases are discussed, namely, those of a uniform or sine disturbance indefinitely applied. These are special cases of the first two disturbances.

The systems will be assumed initially at rest. Pulses will be applied at $t = 0$.

6. The Step Pulse.

This represents the simplest case of a dynamic loading. A disturbance is instantly applied at the time $t = 0$, is kept constant for a definite length of time and is removed at the time $t = t_1$. This disturbance (Figure 3) is, of course, an idealization.



STEP PULSE

FIGURE 3

It never occurs in reality, altho it is perhaps approached in certain types of shock loadings. By reference to the Appendix we have the following responses:

(*a*) During Pulse ($0 \leqq t \leqq t_1$)

$$X = 1 - \cos \frac{2\pi t}{T} \tag{6}$$

(*b*) After Pulse ($t > t_1$)

$$X = 2 \sin \frac{2\pi}{T}\left(t - \frac{t_1}{2}\right) \sin \frac{\pi t_1}{T} \tag{7}$$

Let us consider these responses. It will be noted that both are bounded. The response during the pulse oscillates about a mean position having the value of unity (corresponding to a steady force) attaining peak values of 2 and minimum values of 0. This response depends solely on the natural period of vibration of the system, $T = \frac{2\pi}{a}$. The response subsequent to the pulse oscillates between the positive and negative values of $2 \sin \frac{\pi t_1}{T}$ whose extrema are ± 2 and depends not only upon the natural period of vibration of the system but also upon the instant of removal of the pulse, or rather it depends upon the ratio of these two quantities.

Suppose we analyze a particular case of response. Assume that the duration of the pulse is one-fourth the natural period of vibration of the system, i.e., $t_1 = \frac{1}{4} T$. During the pulse the response follows the law $X = 1 - \cos \frac{2\pi t}{T}$ and attains the maximum value $X = 1$ at the instant of removal of the pulse (Figure 4). Subsequently to this the response is given by

$$X = 2 \sin \frac{2\pi}{T}\left(t - \frac{T}{8}\right) \sin \frac{\pi}{4}$$

which has the first maximum value of 1.41 at $t = \frac{3}{8} T$ (corresponding to the instant when $\sin \frac{2\pi}{T}\left(t - \frac{T}{8}\right) = 1$). This greater of the two maxima is termed "dynamic load factor" and is herein denoted by the symbol n. It represents the absolute maximum value of the response either during or (as in this case) subsequent to the application of a pulse. The dynamic load factor is the coefficient that must be determined in all analysis of systems subjected to dynamic loadings.

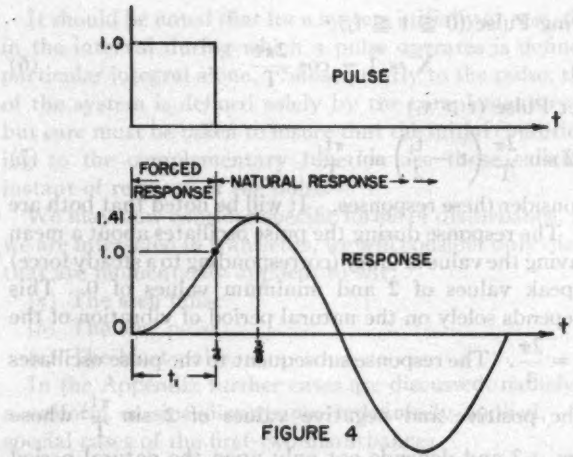


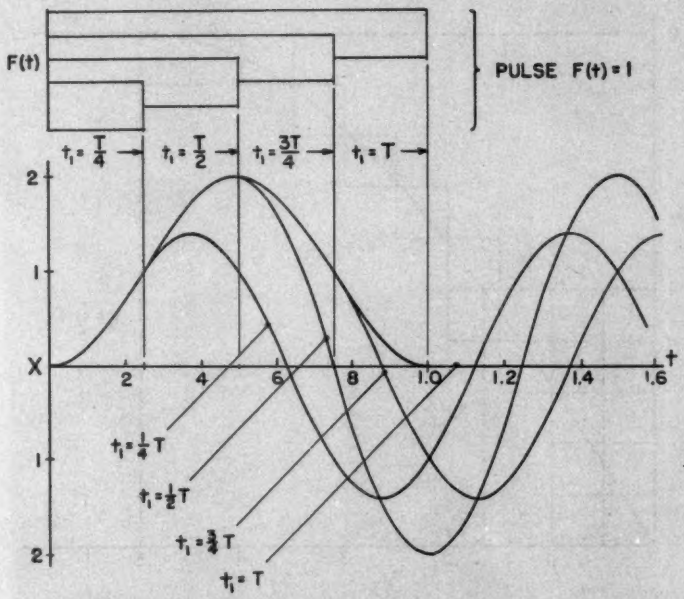
FIGURE 4

The explanation as to why the system may exhibit the greater response after removal of the pulse is simple. During its operation a disturbance imparts a velocity to the system. If, at the instant the disturbance is removed, the velocity of the system is in the direction tending to increase the displacement, then the response will be greater after the pulse because further displacement is necessary to absorb this velocity. If the velocity is in the opposite direction, then the response after the pulse will not be as great.

The above reasoning may be repeated for other values of $\frac{t_1}{T}$. Results for certain other values of this parameter are plotted in Figure 5. Inspection of this figure reveals that:

(*a*) The maximum value of the response and, therefore, of the dynamic load factor increases until $t_1 = \frac{T}{2}$. For values of $t_1 > \frac{T}{2}$ the dynamic load factor, n, equals 2 and is independent of the duration of the pulse.

(*b*) For lesser values of $\frac{t_1}{T}$ the dynamic load factor is given by $n = 2 \sin \frac{\pi t_1}{T}$.



t = DURATION OF PULSE
T = NATURAL PERIOD OF VIBRATION OF SYSTEM
F(t) = DISTURBANCE
X = RESPONSE

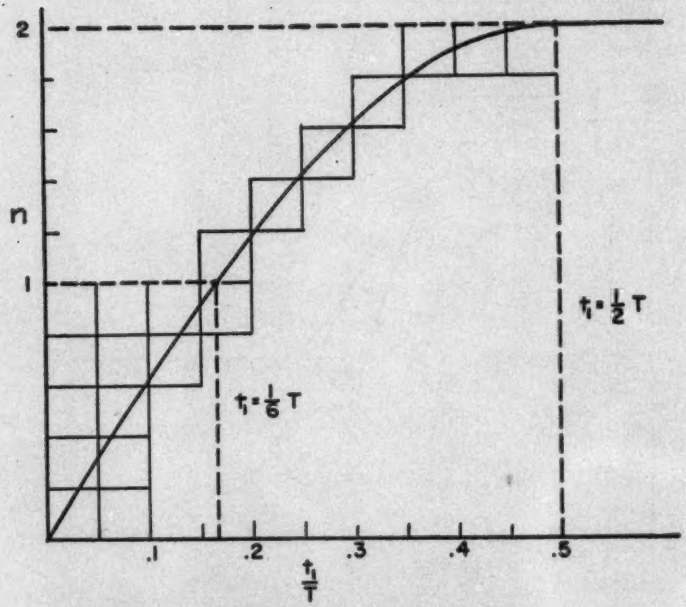
RESPONSE OF A SYSTEM OF ONE DEGREE OF FREEDOM TO A STEP PULSE

FIGURE 5

(*c*) An interesting case presents itself when $t = T$. When this occurs the system is left undisturbed upon removal of the pulse.

The foregoing information on dynamic load factors is presented graphically as in Figure 6. The value for which $n = 1$ i.e., when a dynamic load can be treated as a static load results when

$2 \sin \frac{\pi t_1}{T} = 1$ i.e., when $\frac{t_1}{T} = \frac{1}{6}$.



n = DYNAMIC LOAD FACTOR
t_1 = DURATION OF PULSE
T = NATURAL PERIOD OF VIBRATION OF SYSTEM

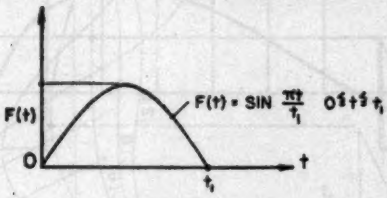
DYNAMIC LOAD FACTOR OF A SYSTEM OF ONE DEGREE OF FREEDOM SUBJECTED TO A STEP PULSE

FIGURE 6

For all durations of t_1 greater than $\frac{T}{6}$ the response of the system to a dynamic load will be greater than for the static load of the same magnitude. On the other hand for all durations of t_1 less than $\frac{T}{6}$ the system will not respond as much.

7. The Sine Pulse.

The commonest form of a disturbance is, perhaps, approximated by a pulse whose form is that of a half sine wave. (Figure 7). The disturbance is applied at $t = 0$ reaches its maximum



SINE PULSE

FIGURE 7

value at $t = \frac{t_1}{2}$ and dies off at $t = t_1$. There is some further merit in studying this form of disturbance, for, since the system is assumed to be linear, other forms of disturbance can be obtained by the simple superposition of sine functions obtained by use of Fourier Series. The disturbance is, then, defined by

$$F(t) = \sin \frac{\pi t}{t_1} \tag{8}$$

and, again, by reference to the Appendix, we have the following responses:

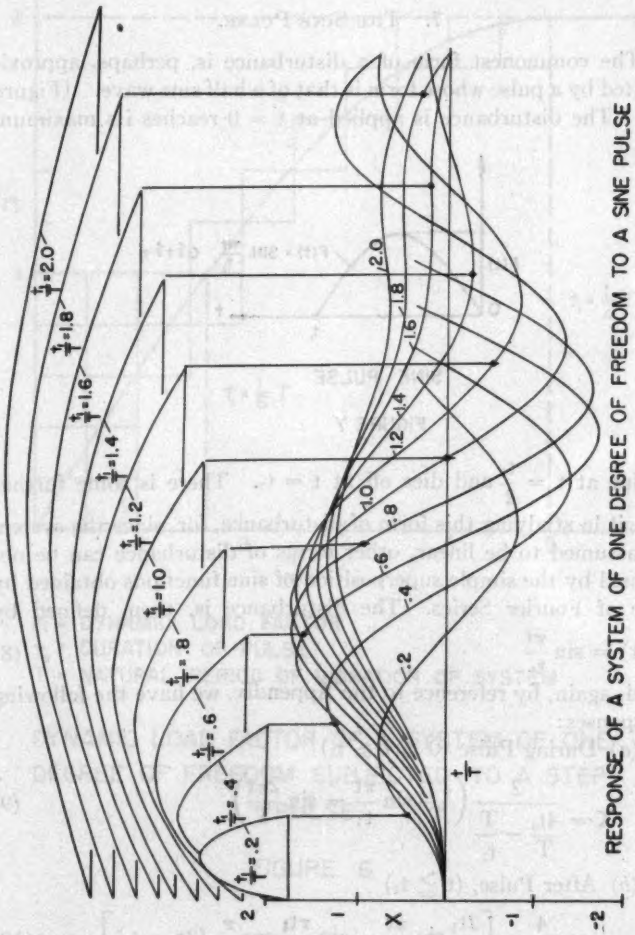
(*a*) During Pulse ($0 \leqq t \leqq t_1$)

$$X = \frac{2}{\frac{4t_1}{T} - \frac{T}{t_1}} \left(\frac{2\,t_1}{T} \sin \frac{\pi t}{t_1} - \sin \frac{2\pi t}{T} \right) \tag{9}$$

(*b*) After Pulse, ($t \geqq t_1$)

$$X = \frac{4}{\frac{4t_1}{T} - \frac{T}{t_1}} \left[\frac{2t_1}{T} \sin \frac{\pi t}{t_1} - \sin \frac{\pi t_1}{T} \cos \frac{\pi}{T} (2t - t_1) \right] \tag{10}$$

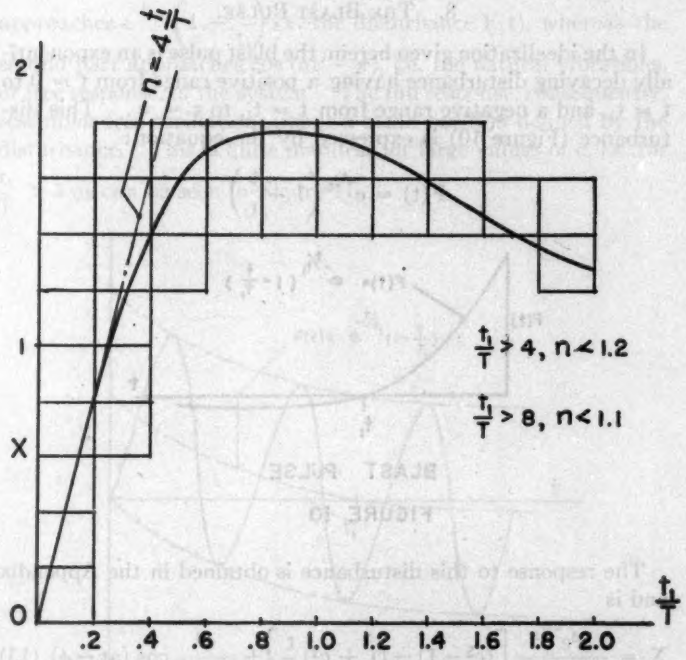
The response to a sine pulse exhibits, in general, the same character as for a rectangular pulse (Figure 8). The peak values attained, however, are not as high. This is quite apparent, for the total impulse imparted by a sine pulse is, evidently, less than



RESPONSE OF A SYSTEM OF ONE DEGREE OF FREEDOM TO A SINE PULSE

FIGURE 8

that imparted by a step pulse. The curve of dynamic load factors, nevertheless, does depart distinctly from the previous one, (Figure 9). After attaining the maximum value n = 1.75 for $t_1 = .8\,T$, the curve drops off for higher values of $\frac{t_1}{T}$. The explanation for this lies in the increasing relative slowness with



DYNAMIC LOAD FACTOR FOR A SYSTEM OF ONE DEGREE OF FREEDOM SUBJECTED TO A SINE PULSE

FIGURE 9

which the pulse is applied for the higher values of $\frac{t_1}{T}$.

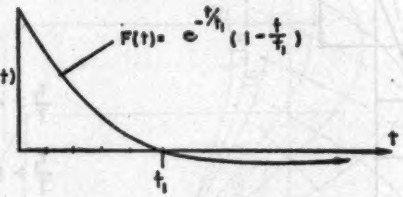
The dynamic load factor no longer has a simple expression, but for values of $\frac{t_1}{T}$ less than .25, it can be represented with sufficient accuracy by

$$n = 4\,\frac{t_1}{T} \tag{11}$$

8. The Blast Pulse.

In the idealization given herein, the blast pulse is an exponentially decaying disturbance having a positive range from $t = 0$ to $t = t_1$, and a negative range from $t = t_1$, to $t = \infty$. This disturbance (Figure 10) is expressed by the equation:

$$F(t) = e^{-t/t_1}\left(1 - \frac{t}{t_1}\right)$$



BLAST PULSE

FIGURE 10

The response to this disturbance is obtained in the Appendix and is

$$X = \frac{c^2 e^{-t/t_1}}{(1+c^2)^2}\left[(c^2-1) - (1+c^2)\frac{t}{t_1}\right] + \frac{c^2}{1+c^2}\cos(at-\phi) \quad (13)$$

where

$$\phi = \sin^{-1}\frac{2c}{1+c^2} \text{ (second quadrant)}$$

$$c = \frac{2\pi t_1}{T}$$

The response is thus made up of 2 parts

(*a*) A decaying forced behavior (particular integral),

$$\frac{c^2 e^{-t/t_1}}{(1+c^2)^2}\left[(c^2-1) - (1+c^2)\frac{t}{t_1}\right]$$

(*b*) A quasi-free oscillation $\frac{c^2}{1+c^2}\cos(at-\phi)$

For large values of c, (i.e., a long duration of pulse relative to the natural period of vibration of the system), the first part

approaches $e^{-t/t_1}\left(1 - \frac{t}{t_1}\right)$ i.e. the disturbance F(t), whereas the second part approaches cos (at $-\pi$), i.e. the natural vibrations, or free response, of the system. The full response, consequently, resembles free oscillations about a mean position defined by the disturbance. This is quite manifest for large values of c, i.e. for $\frac{t_1}{T} > 1$ as can be seen in Figure 11.

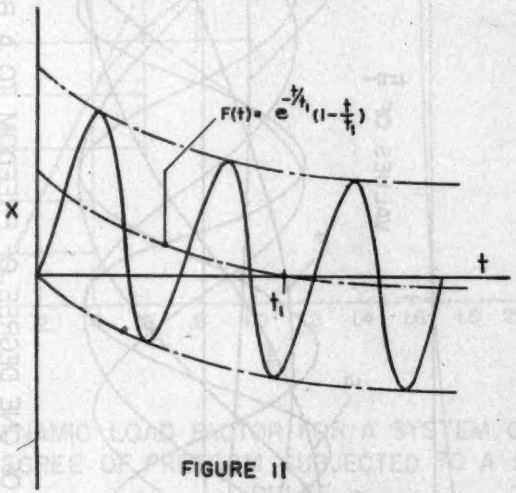


FIGURE 11

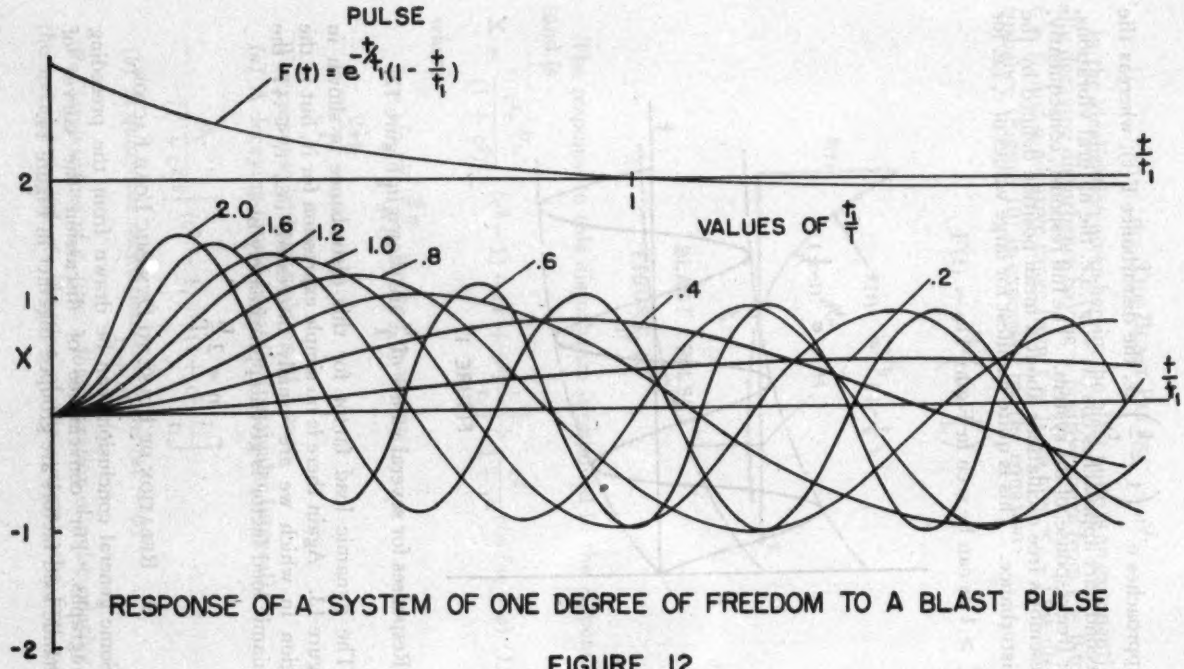
Responses for several values of $\frac{t_1}{T}$ are shown in Figure 12.

The dynamic load factor for this disturbance is shown in Figure 13. Again there is no simple expression for it, but in the region in which we are usually interested, i.e., $t_1 < .4T$ the dynamic load factor is given approximately by

$$n = 2\frac{t_1}{T}$$

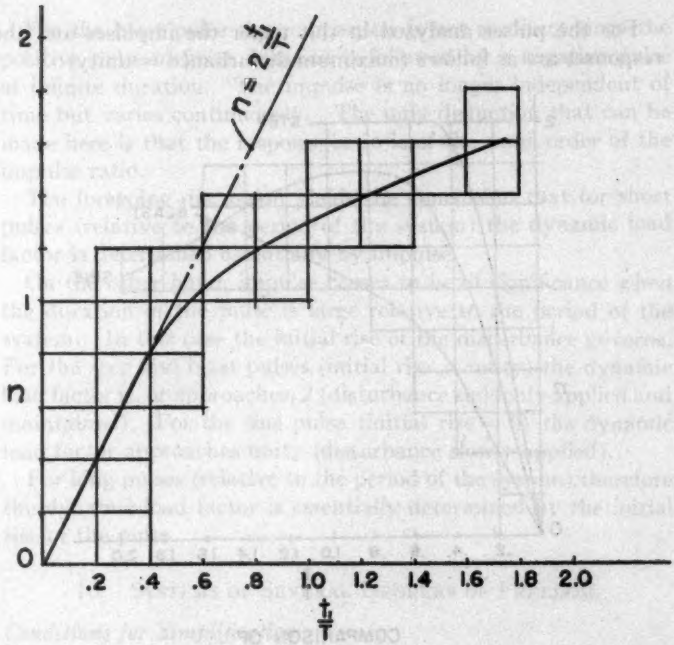
9. Relation of Pulse to Dynamic Load Factor.

Some general conclusions can be drawn from the preceding paragraphs. For convenience of discussion the curves of dynamic load factors are grouped together in Figure 14.



RESPONSE OF A SYSTEM OF ONE DEGREE OF FREEDOM TO A BLAST PULSE

FIGURE 12

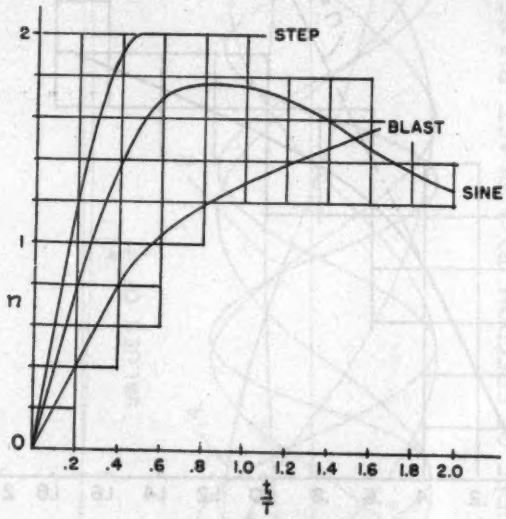


DYNAMIC LOAD FACTOR FOR A SYSTEM OF ONE DEGREE OF FREEDOM SUBJECTED TO A BLAST PULSE

FIGURE 13

Consider first the case in which the duration of the pulse is small compared to the natural period of vibration of the system. Here the positive pulse is applied and removed so rapidly that the system does not respond fully to the disturbance. Since full response requires time, one concludes that the partial response so obtained is a function of the duration of the pulse. This conclusion is corroborated by the linear variation (with respect to pulse duration) of the dynamic load factor in the neighborhood of the origin. This suggests that the dynamic load factors for the different pulses might be compared on the basis of the product of disturbance and time, i.e., impulse.

For the pulses analyzed in this paper the impulses and the responses are as follows (maximum disturbance = unity):



COMPARISON OF
DYNAMIC LOAD FACTORS

FIGURE 14

Pulse	Average Impulse	Ratio (Step = 1)	Response near origin	Ratio (Step = 1)
Step	t_1	1	$2 \sin \frac{\pi t_1}{T} \cong \frac{2\pi t_1}{T}$	1.
Sine	$t_1 \int_0^{t_1} \sin \frac{\pi t_1}{T}\, dt = \frac{2t_1}{\pi}$	.64	$4\frac{t_1}{T}$	.64
Blast*	$\int_0^{t_x} e^{-t/t_1}\left(1-\frac{t}{t_1}\right) = t_x e^{\frac{-t_x}{t_1}}$	.43($t_x = t_1$) .27($t_x = 2t_1$) .15($t_x = 3t_1$)	$2\frac{t_1}{T}$	.32

* Impulse given for time t_x

The comparison for the step and sine pulses is simple and independent of time. The responses for these two pulses are in exact ratio to their impulses.

For the blast pulse the comparison is not so direct since the positive pulse of finite duration is followed by a negative pulse of infinite duration. The impulse is no longer independent of time but varies continuously. The only deduction that can be made here is that the response ratio is of the same order of the impulse ratio.

The foregoing discussion yields the conclusion that for short pulses (relative to the period of the system) the dynamic load factor is determined essentially by impulse.

On the other hand, impulse ceases to be of significance when the duration of the pulse is large relative to the period of the system. In this case the initial rise of the disturbance governs. For the step and blast pulses (initial rise = unity) the dynamic load factor is, or approaches, 2 (disturbance suddenly applied and maintained). For the sine pulse (initial rise = 0) the dynamic load factor approaches unity (disturbance slowly applied).

For long pulses (relative to the period of the system) therefore the dynamic load factor is essentially determined by the initial rise of the pulse.

10. Systems of Several Degrees of Freedom.

Conditions for Simplification.

In article 3 the statement was made that under certain conditions a structure or system possessing several degrees of freedom could be reduced to one of a single degree of freedom without serious error. The conditions under which this can occur will now be discussed without formal proof:

(*a*) The disturbance must be distributed with reasonable uniformity over the structure. This is to insure that only the fundamental mode of vibration of the structure is excited, or that if higher modes are also excited, their contribution will be negligible. When a disturbance exhibits discontinuities in distribution (e.g. a hammer blow on a plate) the higher modes are always present and, indeed, may even become dominant with the result that resemblance is lost between the structure and a system of one degree of freedom.

(*b*) The duration of the pulse must be sufficiently long. This is to insure that the deflected profile of the structure correspond to the disturbance. When the pulse is of extremely short dura-

tion the structure does not have sufficient time to adjust itself to the disturbance. During the period of adjustment transient shock waves, and therefore deformations, are set up so that again similarity is lost to the simple system.

(*c*) There should be no coupling between the fundamental mode of vibration of the structure and the higher modes i.e., each mode of vibration should be independent of the others. This is quite evident. The previous conditions insured that higher modes would not be excited by the externally applied pulse, this condition is to insure that they not be excited by the response of the structure itself.

11. Systems of Several Degrees of Freedom.

Approximate Solution.

If the conditions for which a system of several degrees of freedom can be reduced to one of a single degree of freedom are satisfied, the procedure for converting the dynamic problem to one of statics is as follows:

(*a*) Obtain the fundamental period of free vibration of the structure, T.

(*b*) Determine the character and duration, t_1, of the pulse.

(*c*) Refer to Figures 6, 9, or 13, for the dynamic load factor, n, corresponding to the ratio $\frac{t_1}{T}$ and to the type of pulse. For types of pulses not shown the dynamic load factor can be estimated by interpolation. In doing so the conclusions of article 9 should be kept in mind.

(*d*) The product of the peak dynamic load by the dynamic load factor gives the equivalent static load.

This procedure will now be illustrated by two examples.

12. Flight Deck Structure with Aircraft Landing.

The flight deck structure consists of 3″ × 6″ Douglas Fir planking laid flat athwartships over continuous spans of 24″. The plane tire is such that the length of contract area under maximum load is 24″. At the instant of contact the plane has a velocity of 130 ft. per sec. (Figure 15).

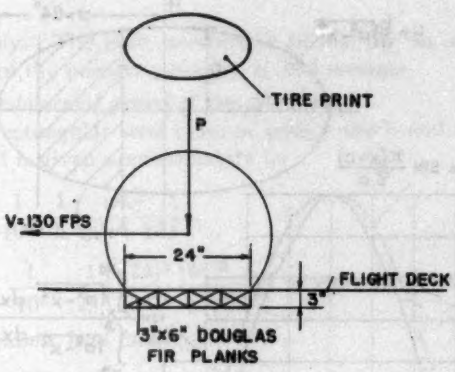


FIGURE 15

(*a*) *Fundamental period of free vibration.*

The fundamental period of free vibration of a beam resting on several equidistant supports is the same as that of a beam hinged at each support. Thus

$$T = \frac{2l^2}{\pi}\sqrt{\frac{\mu}{EI}}$$

l = span = 24″

μ = mass per unit length =

$$\frac{(3'')(6'')(.278)}{382} = .013 \text{ lbs. sec}^2 \text{ per sq. in.}$$

(2.78 lbs. per cu. in. = unit weight of Douglas Fir.
382 in. per sec^2 = g = acceleration of gravity)

E = modulus of elasticity = 1.6×10^6 lbs. per sq. in. for Douglas Fir.

I = moment of inertia = 13.5 in^4.

$$T = \frac{2(24)^2}{\pi}\sqrt{\frac{.013}{(1.6 \times 10^6)(13.5)}} = .0091 \text{ sec.}$$

(*b*) *Character and duration of pulse.*

The tire print approximates an ellipse. For such a print the pulse on the plank generated by the tire rolling over it is given in Figure 16 and approaches closely a sine curve. The dynamic

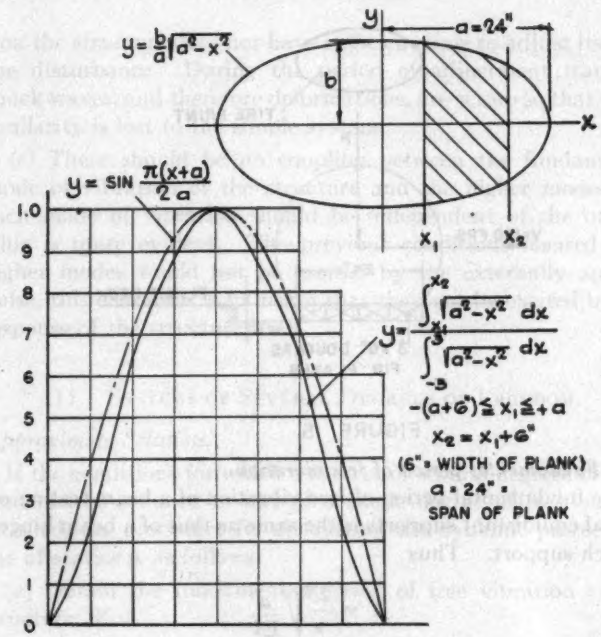


FIGURE 16

load factor is consequently given by Figure 9.

For a plane velocity of 130 ft. per sec. the time it takes for the tire to roll over the plank is

$$t_1 = \frac{2}{130} = .016 \text{ sec.}$$

(*c*) *Dynamic load factor.* $\frac{t_1}{T} = \frac{.016}{.0091} = 1.75$

From Figure 9, n = 1.38.

Thus in this case the landing load of the plane must be increased by 38% before it can be applied as a static load to the flight deck structure!

13. Deck Structure Subjected to Gun Blast.

The deck structure consists of panels of plating .25 in. thick resting on crossing supports spaced 24 in. and 48 in. apart

respectively. The peak pressure is 20 lbs. per sq. in. and the duration of the positive pulse, t_1, is .002 seconds.

(*a*) *Fundamental period of free vibration.*

For a rectangular steel plate of sides a and b and thickness h the period is given approximately by

$$T \cong \frac{1}{10^5} \cdot \frac{1}{h}\left(\frac{a^2b^2}{a^2+b^2}\right)$$

$$= \frac{1}{10^5} \cdot \frac{1}{.25}\left[\frac{(24)^2\,(48)^2}{(24)^2+(48)^2}\right] = .0183 \text{ sec.}$$

(*b*) *Dynamic load factor*

$$\frac{t_1}{T} = \frac{.002}{.0183} = .11$$

Referring to Figure 13, n = .22. The equivalent static load is, consequently, (.22) (20) or 4.4 lb. per sq. in. As against the finding of the previous article, in this case the blast load is to be reduced to only 22% of its peak value before it is applied as a static load to the design of the deck structure.

14. Conclusions.

The response of a structure subjected to a dynamic load of peak value P_o may differ materially from that of the same structure acted upon by the corresponding static load of constant magnitude P_o.

Depending on the ratio of the duration of the positive pulse of the load, t_1, to the fundamental period of free vibration of the structure to which it is applied, T, the strains induced in the structure may be either over-estimated or under-estimated considerably if the dynamic nature of the load is not recognized.

A suitable system of design for dynamic loadings is to determine the dynamic load factor for the specific case and to replace the dynamic loading by an equivalent static loading. If this is not done, material waste may result or serious risks may incur.

Determination of the proper dynamic load factor to use in design requires only knowledge of the character of the external loading and of the natural period of vibration of the structure. In practice, these are either known or can be easily obtained by calculation or from experimental records.

The examples show that the method can be readily handled in the drafting room. Its relative simplicity and theoretical basis should prove it superior to the cabalistic art that sometimes (?) is used in design of structures.

15. Acknowledgement.

The author is indebted to Mr. Mark Kurtz of the Bureau of Ships for many valuable suggestions and for checking the mathematical work contained in the Appendix.

16. References.

(*a*) J. M. Frankland, "Effects of Impact on Simple Elastic Structures". Taylor Model Basin Report No. 481.

This is a valuable report covering a wider field than discussed in this paper. To the author's knowledge, it is the first report giving a good physical picture of the characteristic features of impact phenomena in mechanical systems representative of ship's structures.

(*b*) M. F. Gardner and J. L. Barnes "Transients in Linear Systems". Vol. 1—John Wiley & Sons—New York.

This is an excellent textbook for the solution of differential equations by the Laplace transformation method.

Appendix.

Solution of the Differential Equation $\frac{1}{a^2} \cdot \frac{d^2X}{dt^2} + X = F(t)$

(*a*) *Arbitrary forcing term F(t)*

The direct Laplace transformation is given as

$$L\{X(t)\} = \int_0^\infty X(t)\, e^{-st}\, dt = \overline{X}(s)$$

The inverse Laplace transformation is

$$L^{-1}\{\overline{X}(s)\} = X(t),\ 0 \leqq t$$

and is restricted to the positive range of t.

In our case, rewriting the equation in operational form, we have

$$(s^2 + a^2)\, X = a^2\, F(t)$$

where $s = \frac{d}{dt}$, $s^2 = \frac{d^2}{dt^2}$ etc., and the direct transformation is

$$L\{X(t)\} = (s^2+a^2)\int_o^\infty X(t)e^{-st} = sX_o + \left(\frac{dx}{dt}\right)_o + a^2\int_o^\infty F(t)e^{-st}\,dt$$

where the subscript 0 indicates that the values are taken at $t = 0$. Letting

$$\overline{X}(s) = \int_o^\infty X(t)\,e^{-st}\,dt, \quad \overline{F}(s) = \int_o^\infty F(t)\,e^{-st}\,dt$$

we then have

$$\overline{X}(s) = \frac{sX_o + \left(\frac{dX}{dt}\right)_o}{s^2+a^2} + \frac{a^2\,\overline{F}(s)}{s^2+a^2}$$

The inverse transforms are as follows

$$L^{-1}\left\{\frac{s}{s^2+a^2}\right\} = \cos at$$

$$L^{-1}\left\{\frac{1}{s^2+a^2}\right\} = \frac{1}{a}\sin at$$

$$L^{-1}\left\{\frac{a^2\,\overline{F}(s)}{s^2+a^2}\right\} = L_1^{-1}\left\{a^2\,\overline{F}(s)\right\} \cdot L_2^{-1}\left\{\frac{1}{s^2+a^2}\right\} = a^2\int_o^t f_1(t)\cdot f_2(t-\tau)\,d\tau$$

where

$$f_1(t) = L_1^{-1}\left\{a^2\,F(s)\right\} = a^2\,F(t)$$

$$f_2(t-\tau) = L_2^{-1}\left\{\frac{1}{s^2+a^2}\right\}_{(t-\tau)} = \frac{1}{a}\sin a\,(t-\tau)$$

and τ is the variable of integration to be distinguished from the limit of integration t.

Therefore

$$L^{-1}\left\{\frac{a^2\,\overline{F}(s)}{s^2+a^2}\right\} = a\int_o^t F(t)\sin a\,(t-\tau)\,d\tau$$

The response for an arbitrary forcing term is consequently

$$X(t) = X_o\cos at + \frac{1}{a}\left(\frac{dX}{dt}\right)_o \sin at + a\int_o^t F(t)\sin a(t-\tau)\,d\tau \quad (a)$$

(*b*) *Forcing term absent*, $F(t) = 0$.

When a disturbance has ceased to operate the solution is then simply

$$X(t) = X_o \cos at + \frac{1}{a}\left(\frac{dX}{dt}\right)_o \sin at \qquad (b)$$

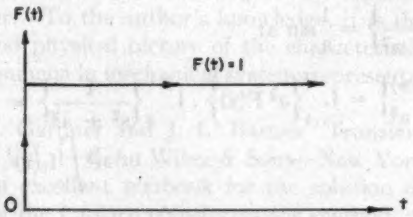
If instead of assuming initial conditions at $t = 0$, they are taken at some other time $t = t_1$, we have, obviously

$$X(t) = X_1 \cos a(t - t_1) + \frac{1}{a}\left(\frac{dX}{dt}\right)_1 \sin a\,(t - t_1),\ t \geq t_1 \qquad (c)$$

This response is periodic of period $T = \frac{2\pi}{a}$. This is the natural period of vibration of the system.

(*c*) *Unit Disturbance*, $F(t) = 1$.

If a disturbance is instantly applied and held indefinitely, $F(t) = 1$ (Figure 17). For initial conditions $X_o = \left(\frac{dX}{dt}\right) = 0$, the



UNIT DISTURBANCE

FIGURE 17

direct transformation is given by

$$(s^2 + a^2)\int_o^\infty X(t)e^{-st}\,dt = a^2 \int_o^\infty e^{-st}\,dt$$

Here

$$\overline{F}(s) = \frac{1}{s}$$

and

$$\overline{X}(s) = \frac{1}{s(s^2 + a^2)}$$

The inverse transform is

$$L^{-1}\left\{\frac{1}{s(s^2 + a^2)}\right\} = \frac{1}{a^2}\,(1 - \cos at)$$

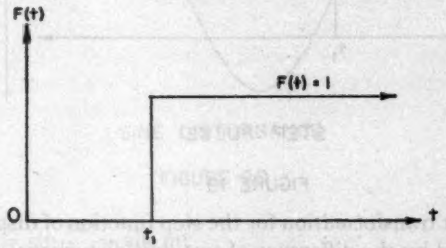
yielding

$$X(t) = (1 - \cos at) = \left(1 - \cos \frac{2\pi t}{T}\right) \qquad (d)$$

In this case the same solution can, of course, be obtained by simple integration of equation (a), since for the specified conditions

$$X(t) = a\int_0^t \sin a(t - \tau)\, d\tau = (1 - \cos at) = \left(1 - \cos \frac{2\pi t}{T}\right) (d)$$

If instead of assuming initial conditions at $t = 0$ they are taken at $t = t_1$ (Figure 18), the direct transformation is



UNIT DISTURBANCE
ORIGIN AT t_1

FIGURE 18

$$\bar{F}(s) = \frac{1}{s} e^{-st_1}$$

Use will be made of this in the next case.

(*d*) *Step Pulse.*

Here

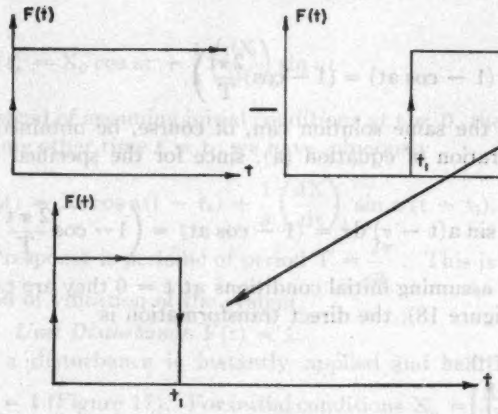
$F(t) = 1, 0 \leqq t \leqq t_1$

$F(t) = 0, t \geqq t_1$

where t_1 is the duration of the pulse (Figure 19).

For initial conditions $X_o = \left(\frac{dX}{dt}\right)_o = 0$ the direct transformation of the equation is

$$(s^2 + a^2)\int_0^\infty X(t)\, e^{-st}\, dt = a^2 \int_0^\infty F(t)\, e^{-st}\, dt$$



STEP PULSE

FIGURE 19

The direct transformation for the step function of displacement t_1 is obtained by the difference of two unit disturbances, the first beginning at $t = 0$, the second at $t = t_1$, both extending to infinity. By reference to the previous case

$$\bar{F}(s) = \frac{1}{s}(1 - e^{-st_1})$$

Giving

$$\bar{X}(s) = \frac{1}{s(s^2 + a^2)} - \frac{e^{-st_1}}{s(s^2 + a^2)}$$

The inverse transforms are

$$L^{-1}\left\{\frac{1}{s(s^2 + a^2)}\right\} = \frac{1}{a^2}(1 - \cos at),\ t \leqq t_1$$

$$L^{-1}\left\{\frac{e^{-st_1}}{s(s^2 + a^2)}\right\} = \frac{1}{a^2}\left[1 - \cos a(t - t_1)\right],\ t \geqq t_1$$

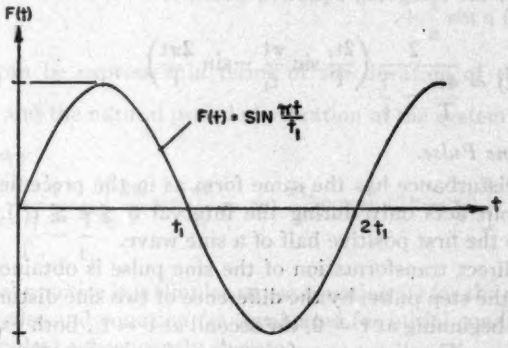
Consequently

$$X(t) = (1 - \cos at) - \left[1 - \cos(t - t_1)\right] = 2 \sin a\left(t - \frac{t_1}{2}\right) \sin \frac{at_1}{2}$$

$$= 2 \sin \frac{2\pi}{T}\left(t - \frac{t_1}{2}\right) \sin \frac{\pi t_1}{T} \qquad (e)$$

(*e*) *Sine Disturbance.*

The disturbance has the form (Figure 20).



SINE DISTURBANCE

FIGURE 20

$$F(t) = \sin \frac{\pi t}{t_1} = \sin mt$$

and the direct transformation of the equation is

$$(s^2 + a^2) \int_0^\infty X(t)\, e^{-st}\, dt = a^2 \int_0^\infty e^{-st} \sin mt\, dt$$

From which

$$\overline{F}(s) = a^2 \int_0^\infty e^{-st} \sin mt\, dt = \frac{a^2 m}{s^2 + m^2}$$

and

$$\overline{X}(s) = \frac{a^2 m}{(s^2 + m^2)(s^2 + a^2)} = \frac{a^2 m}{a^2 - m^2}\left(\frac{1}{s^2 + m^2} - \frac{1}{s^2 + a^2}\right)$$

Since $L^{-1}\left\{\frac{1}{s^2 + m^2}\right\} = \frac{1}{m} \sin mt$ etc., the inverse transformation yields

$$X(t) = \frac{a^2 m}{a^2 - m^2}\left(\frac{1}{m} \sin mt - \frac{1}{a} \sin at\right)$$

In terms of the duration of the half period of the disturbance,

$t_1 = \frac{\pi}{m}$ and the natural period of vibration of the system, $T = \frac{2\pi}{a}$, the foregoing equation becomes

$$X(t) = \frac{2}{\frac{4t_1}{T} - \frac{T}{t_1}} \left(\frac{2t_1}{T} \sin \frac{\pi t}{t_1} - \sin \frac{2\pi t}{T} \right) \qquad (f)$$

(*f*) *Sine Pulse.*

The disturbance has the same form as in the preceding paragraph, but acts only during the interval $0 \leqq t \leqq t_1$ i.e., it is equal to the first positive half of a sine wave.

The direct transformation of the sine pulse is obtained, similarly to the step pulse, by the difference of two sine disturbances, the first beginning at $t = 0$, the second at $t = t_1$, both extending to infinity. The direct transformation of a sine function starting at $t = t_1$ is

$$\overline{F}(s) = \frac{a^2 m}{s^2 + m^2} e^{-st_1}$$

we consequently have for the single half wave of the sine pulse.

$$\overline{F}(s) = \frac{a^2 m}{s^2 + m^2} (1 - e^{-st_1})$$

and therefore

$$\overline{X}(s) = \frac{a^2 m}{(s^2 + m^2)(s^2 + a^2)} (1 - e^{-st_1})$$

The inverse transforms are

$$L^{-1} \left\{ \frac{a^2 m}{(s^2 + m^2)(s^2 + a^2)} \right\} = \frac{a^2 m}{a^2 - m^2} \left(1 \sin mt - \frac{1}{a} \sin at \right), \; t \leqq t_1$$

$$L^{-1} \frac{a^2 m e^{-st_1}}{(s^2 + m^2)(s^2 + a^2)} = \frac{a^2 m}{a^2 - m} \left[\frac{1}{m} \sin m(t - t_1) - \frac{1}{a} \sin a(t - t_1) \right], \; t \geqq t_1$$

The response is consequently

$$X(t) = \frac{a^2 m}{a^2 - m^2}\left[\frac{1}{m}\sin mt - \frac{1}{a}\sin at - \frac{1}{m}\sin m(t - t_1) + \frac{1}{a}\sin a(t - t_1)\right]$$

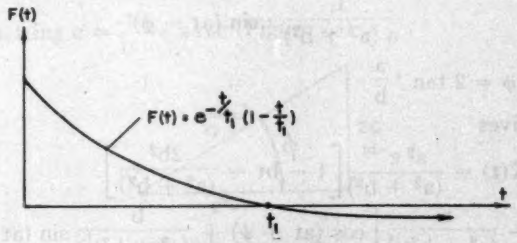
which can be expressed in terms of the duration of the pulse $t_1 = \frac{\pi}{m}$ and the natural period of vibration of the system $T = \frac{2\pi}{a}$ as follows:

$$X(t) = \frac{4}{\frac{4t_1}{T} - \frac{T}{t_1}}\left[\frac{2t_1}{T}\sin\frac{\pi t}{t_1} - \sin\frac{\pi t_1}{T}\cos\frac{\pi}{T}(2t - t_1)\right] \qquad (g)$$

In actual practice it is simpler to use equation (f) for the duration of the pulse and equation (c) (evaluated for initial conditions at end of pulse) subsequently thereto.

(*g*) *Blast Disturbance.*

The disturbance has the form (Figure 21)



BLAST DISTURBANCE

FIGURE 21

$$F(t) = e^{-t/t_1}\left(1 - \frac{t}{t_1}\right)$$

and we have for the direct transformation

$$(s^2 + a^2)\int_0^\infty X(t)\, e^{-st}\, dt = a^2\int_0^\infty e^{-st}.\, e^{-bt}\,(1 - bt)\, dt$$

where for convenience $b = \frac{1}{t_1}$

$$\overline{F}(s) = a^2\int_0^\infty e^{-(s+b)t}\, dt - a^2 b\int_0^\infty t.\, e^{-(s+b)t}\, dt$$

Now

$$a^2 \int_0^\infty e^{-(s+b)t}\, dt = \frac{a^2}{s+b}$$

$$a^2 \int_0^\infty t \cdot e^{-(s+b)t}\, dt = \frac{a^2 b}{(s+b)^2}$$

Therefore

$$\overline{X}(s) = \frac{a^2}{(s+b)(s^2+a^2)} - \frac{a^2 b}{(s+b)^2 (s^2+a^2)}$$

The inverse transforms are

$$L^{-1}\left\{\frac{1}{(s+b)(s^2+a^2)}\right\} = \frac{e^{-bt}}{(a^2+b^2)} - \frac{1}{a} \cdot \frac{1}{(a^2+b^2)^{\frac{1}{2}}} \cdot \cos(at+\psi)$$

where $\psi = \frac{\pi}{2} - \tan^{-1}\frac{a}{b}$

$$L^{-1}\left\{\frac{1}{(s+b^2)(s^2+a^2)}\right\} = \frac{e^{-bt}}{(a^2+b^2)}\left[t + \frac{2b}{(a^2+b^2)}\right] + \frac{1}{a(a^2+b^2)} \cdot \sin(at-\phi)$$

where $\phi = 2\tan^{-1}\frac{a}{b}$

This gives

$$X(t) = \frac{a^2 e^{-bt}}{(a^2+b^2)}\left[1 - bt - \frac{2b^2}{(a^2+b^2)}\right] - \frac{a}{(a^2+b^2)^{\frac{1}{2}}}\left[\cos(at+\psi) + \frac{b}{(a^2+b^2)^{\frac{1}{2}}}\sin(at-\phi)\right]$$

If we let $c = \frac{a}{b}$, the first right hand term reduces easily to

$$\frac{c^2 e^{-bt}}{(1+c^2)^2}\left[(c^2-1) - (1+c^2)bt\right]$$

In order to reduce the second right hand term we observe that from trigonometry (Figure 22)

$$\cos(at+\psi) = \sin\left(\frac{\phi}{2} - at\right)$$

and since $\frac{a}{(a^2+b^2)^{\frac{1}{2}}} = \sin\frac{\phi}{2}$ and $\frac{b}{(a^2+b^2)^{\frac{1}{2}}} = \cos\frac{\phi}{2}$

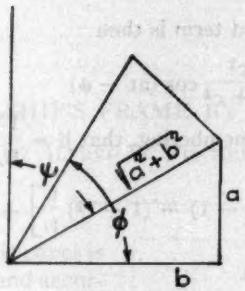


FIGURE 22

we have by these substitutions

$$-\sin\frac{\phi}{2}\left[\sin\left(\frac{\phi}{2}-at\right)+\cos\frac{\phi}{2}(at-\phi)\right]$$

which upon further trigonometric reduction becomes

$$\sin^2\frac{\phi}{2}\cos(at-\phi) \quad \text{or} \quad \frac{a^2}{(a^2+b^2)}\cos(at-\phi)$$

Again, letting $c=\frac{a}{b}$ we have (Figure 23)

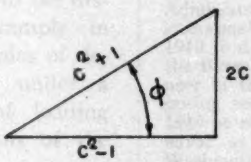


FIGURE 23

$$\frac{a^2}{a^2+b^2}=\frac{c^2}{1+c^2}$$

$$\sin\frac{\phi}{2}=\frac{a}{(a^2+b^2)^{1/2}}=\frac{c}{(1+c^2)^{1/2}}$$

$$\cos\frac{\phi}{2}=\frac{1}{(1+c^2)^{1/2}}$$

$$\sin\phi=2\sin\frac{\phi}{2}\cos\frac{\phi}{2}=\frac{2c}{1+c^2}$$

$$\cos\phi=\frac{c^2-1}{1+c^2}$$

The second right hand term is then

$$\frac{c^2}{1+c^2}\cos(at-\phi)$$

and the response $\left(\text{remembering, that } b=\frac{1}{t_1}\right)$

$$X(t)=\frac{c^2 e^{-t/t_1}}{(1+c^2)^2}\left[(c^2-1)-(1+c^2)\frac{t}{t_1}\right]+\frac{c^2}{1+c^2}\cos(at-\phi)$$

where

$$\phi=\sin^{-1}\frac{2c}{1+c^2}$$

and

$$c=\frac{a}{b}=\frac{2\pi t_1}{T}$$

ANALYSIS OF A SHIP'S FRAME BY SUPERPOSITION.

By Alfred W. Fischer.

The purpose of this paper is to present a simple and accurate method for calculating the moments, reactions, and thrusts in a multiple deck frame composed of straight or curved members having a constant or non-uniform cross-section. The subject of designing transverse frames, in the wide sense of the term, "design", is not treated; the paper is restricted to the discussion of an example in engineering mechanics of determining stresses under a given condition of loading when all dimensions of the frame are known.

The analysis is based on the column analogy[1] and the moment distribution method,[2] but their application to a ship's frame is new.

The Author.

Alfred W. Fischer was born in Trenton, Illinois, educated in the public schools there, and received a degree in Civil Engineering from Valparaiso University in 1909. After graduation and until 1918 he worked on drainage and construction work. Three years of this period was spent in Canada. He served in World War I with one year's service abroad. From 1919 to 1936 he worked as a structural engineer on many building and bridge projects. From 1936 to 1940 he worked for the Veterans Administration as an earthquake stress analyst on hospitals. From 1940 to date he has worked for the Bureau of Ships as an engineer in the stability and computing section. In the fall of 1940 he received his training in naval architecture at George Washington University. During his engineering career he has written numerous articles and discussions which have appeared in various engineering periodicals.

Preliminary Considerations.

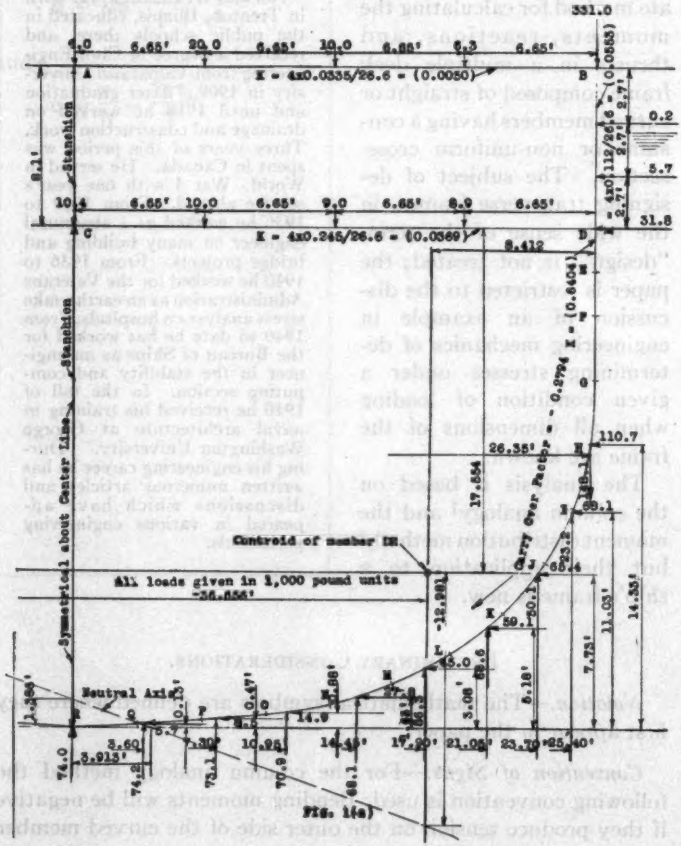
Notation.—The mathematical symbols are defined where they first appear in the paper.

Convention of Signs.—For the column analogy method the following convention is used: Bending moments will be negative if they produce tension on the outer side of the curved member

and positive if they produce compression on the outer side. Coordinates will be taken as positive when measured upward or to the right from the axes.

For moment distribution the resisting moment M_{AB} for member AB and the fixed-end moment FM_{AB} are positive when they act in a clockwise direction and negative when they act in a counterclockwise direction.

Length of Members.—In the solution of the example, the length L is taken between neutral axis intersections of the members.



Fixed-End Moments.—FM_{AB} for member AB is the moment at end A of a member in which rotation is prevented at both ends, that is, in which the angle change is zero at both ends.

Stiffness Factor.—The stiffness factor is defined as the moment required to rotate a simply supported end of a member through a unit angle when the other end is fixed. For prismatic members, $K = 4EI/L$, where I = moment of inertia of the member, and E = modulus of elasticity. Since E is constant for all members, it can be called unity, then $K = 4I/L$. K for the curved member DR in Figure 1(a) at end D = 0.6404. Method of determination given later.

Carry Over Factor.—The carry over factor C is the ratio of moment induced at a fixed-end to moment producing rotation at the other end. $C = \frac{1}{2}$ for prismatic members. C for the curved member DR from D to R equals -0.2794. Method of determination given later.

Outline of the Method of Procedure—The proposed method may be divided into the following steps:

Step 1. Sketches showing all possible conditions of deformation of the frame are made. One of the sketches will show the undeflected frame with the loads that may be acting, and the others will show the frame for each unknown linear joint displacement.

Step 2. Fixed-end moments set up by the vertical and horizontal loads and the unknown linear joint displacement of the members before rotation of the joints is permitted, are determined and recorded on the appropriate sketches of step 1.

Step 3. The joints are balanced for each condition of deformation in step 1 by the moment distribution method, and the final moments are shown on the deformation sketches. For the curved members the balancing moments must be recorded separately from the fixed-end moments. This arrangement is necessary as the change in the horizontal thrust and the vertical reaction due to the rotation of the joints must be calculated.

Step 4. Equilibrium equations are written for the frame with the given loads; and, after proper substitutions, the unknown displacements are evaluated. These substitutions are determined from the consideration of appropriate free bodies on which

the moments in step 3 and all forces are shown acting. The moment at the end of any member in a free body must equal the sum of the moments obtained at the end of that member for several deformation conditions. It will include a numerical value, if the loads act on any member, and additional values that are functions of the unknown linear displacements.

Step 5. With all displacements known the moments in step 3 which are functions of these displacements are evaluated, and the final numerical values of the moments at the ends of all members are determined by summation.

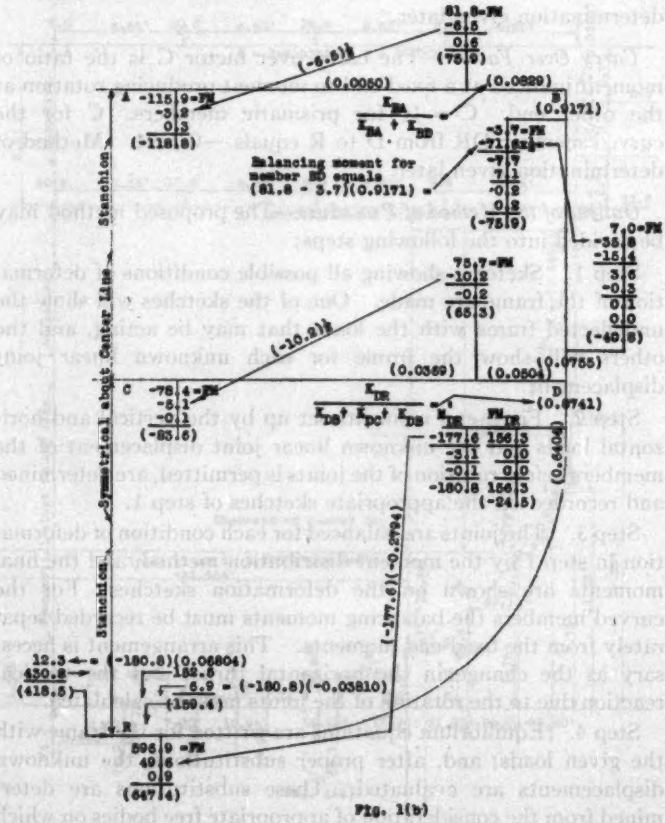


Fig. 1(b)

Typical Example.—Figure 1(a) represents a symmetrical multiple deck frame, subjected to the loads as shown. The K – values of all members are shown in parentheses. It is desired to compute the values of the moments, reactions, and thrusts at the ends of all members.

Step 1. Two possible conditions of deformation exist—one for the loads as shown in Figure 1(a), and the other for the vertical displacement y (assumed to act upward at joints A, C, and R). Since it is assumed that the stanchions AC and CR do not change in length, the displacements y at joints A, C, and R is equal. These are shown in Figures 1(b) and 1(c).

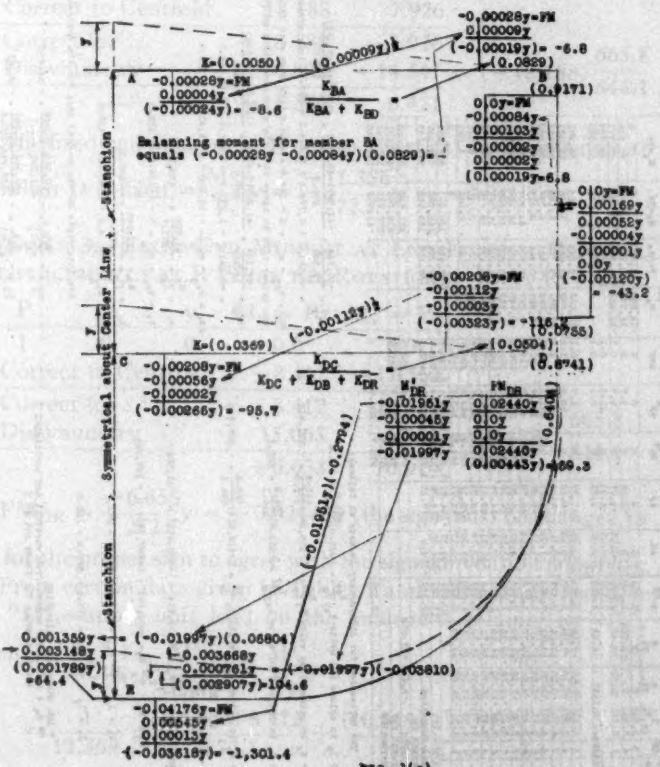
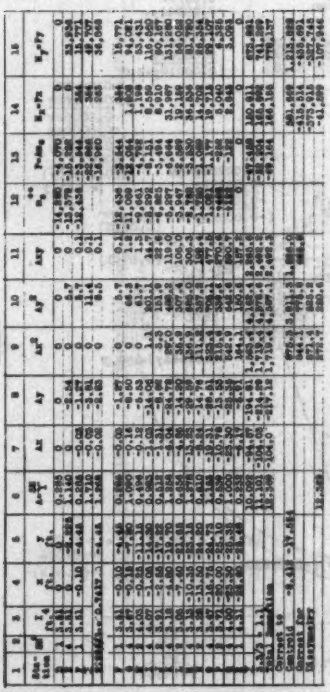


FIG. 1(c)

Step 2. Fixed-end moments are computed for the deformed conditions of Figures 1(b) and 1(c). Fixed-end moments for members AB, BD, and CD for the given loads can be solved by substituting in equations given in many textbooks,[3] these moments are recorded in Figure 1(b). The fixed-end moments for members AB and CD for a vertical displacement y at joints A and C can be calculated by the equations $-6EIy/L^2$ (E = unity), these values are recorded in Figure 1(c).



The fixed-end moments for the curved member DR for the loads as shown are calculated in Table 1 by the column analogy method and are recorded in Figure 1(b). The fixed-end moments for member DR due to a vertical displacement y of joint R when joint D is fixed are recorded in Figure 1(c). See Tables 2 and 3 for the solution of their values by the column analogy method.

TABLE 2.—FIXED-END MOMENT AT END R FOR A VERTICAL DISPLACEMENT Y AT R WHEN END D IS FIXED.

P	x	y	$M_x = Px$	$M_y = Py$	
1	0	0	0.0	0.0	
Correct to Centroid			18.188	7.926	
Correct for			−18.188	−7.926	
Dissymmetry			−6.802	−14.347	$= (-18.188)\frac{665.8}{844.1}$
			−11.386	6.421	

The fixed-end moment at R due to a vertical displacement y at R when D is fixed $= \frac{M'_x}{I'_x}y = \frac{-11.386}{272.7}y = -0.04176y.$

TABLE 3.—FIXED-END MOMENT AT END D FOR A VERTICAL DISPLACEMENT Y AT R WHEN THE ROTATION OF JOINT R = zero.

P	x	y	$M_x = Px$	$M_y = Py$	
1	0	0	0.0	0.0	
Correct to Centroid			−8.412	−17.554	
Correct for			8.412	17.554	
Dissymmetry			15.065	6.635	$= (8.412)\frac{665.8}{844.1}$
			−6.653	10.919	

$FM_{DR} = \frac{-6.653}{272.7}y = -0.02440y$ (the sign must be changed to + for the proper sign to agree with the sign convention adopted).

From certain data given in Tables 1 and 3, for a unit rotation at "D",—apply unit load on the analogous column[4] at "D", then: $m_{i_D} = \frac{1}{A} + \frac{M'_x}{I'_x}x + \frac{M'_y}{I'_y}y$ (1)

$$= \frac{1}{12.369} + \frac{(-6.653)\,8.412}{272.7} + \frac{(10.919)\,17.554}{250.6}$$

$= 0.6404$ = Stiffness Factor at end D for member DR.

For a unit rotation at "R"—apply unit load on analogous column at "R" then:

$$m_{i_R} = \frac{1}{12.369} + \frac{(-6.653)(-18.188)}{272.7} + \frac{10.919(-7.926)}{250.6} = 0.1789$$

Carry Over Factor from end D to end R $= \frac{m_{i_R}}{m_{i_D}} = \frac{0.1789}{0.6404} = 0.2794$. The carry over factor as calculated by the column analogy is positive, but for the moment distribution method as adopted this carry over factor sign is minus.

For a vertical displacement y at R the vertical shear at R $= v_{i_R} = \frac{1}{I'_x}y = \frac{1}{272.4}y = 0.003668y$

In Figure 1(d) summation M about D $= 0$,

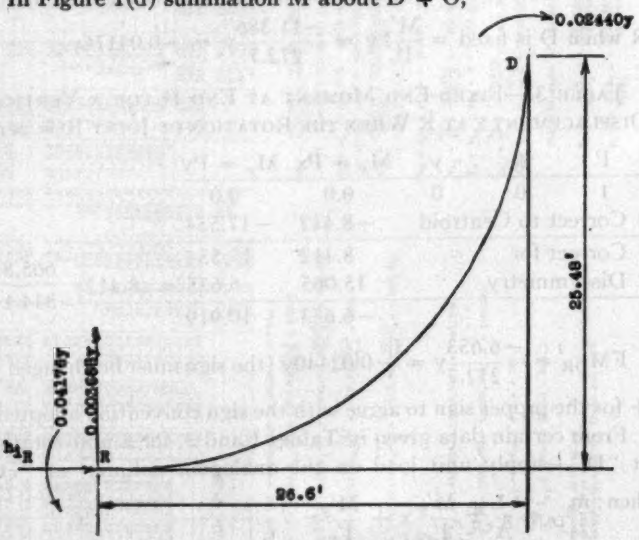


FIG. 1(d)

$-0.04176y + 0.02440y + \frac{1}{2}(0.003668y)(26.6) - 25.48h_{i_R} = 0$;
and $h_{i_R} = 0.003148y$

Step 3. In this example the joints are balanced by the moment distribution method. Figures 1(b) and 1(c) show the distirbutions of the fixed-end moments. The distributions of the fixed-end moments which are recorded in Figures 1(b) and 1(c) are similar to the procedure for frames with straight members except that the corrections M'_{DR} (balanced moments at joint D for the curved member) are recorded in a separate column from the fixed-end moments. This arrangement is necessary as the change in the horizontal thrust h_R and the vertical reaction v_R due to the rotation of the joint D must also be calculated.

From data Table 3 the correction necessary due to rotation of joint D for $h'_{i_R} = \dfrac{M'_y\,(M'_{DR})}{I'_y\,(\text{stiffness factor DR})} = \dfrac{10.919}{250.6\,(0.6404)}$ $(M'_{DR}) = 0.06804M'_{DR}$, and for $v'_{i_R} = \dfrac{-6.653}{272.4\,(0.6404)}\,M'_{DR} = -0.03801M'_{DR}$

Step 4. To determine the value of y an equilibrium equation for the frame must be written.

In Figure 1(e) from summation $V_A + V_C + V_R = O$ (2)

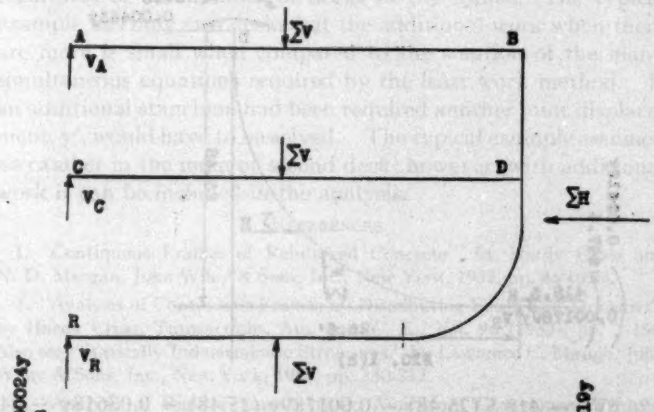


FIG. 1(e)

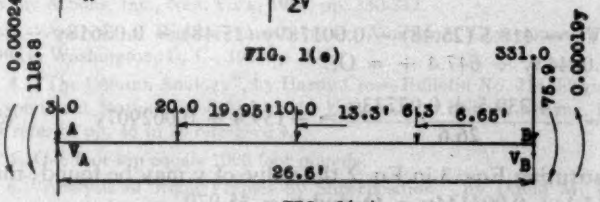


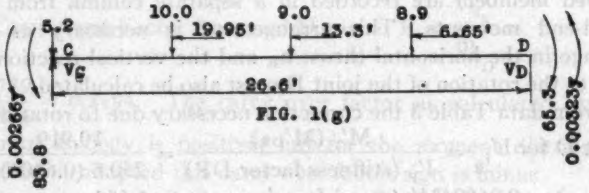
FIG. 1(f)

In Figure 1(f) summation M about B = O,

$$26.6V_A - 0.00024y - 0.00019y - 118.8 + 75.9 - 653.7 = O,$$

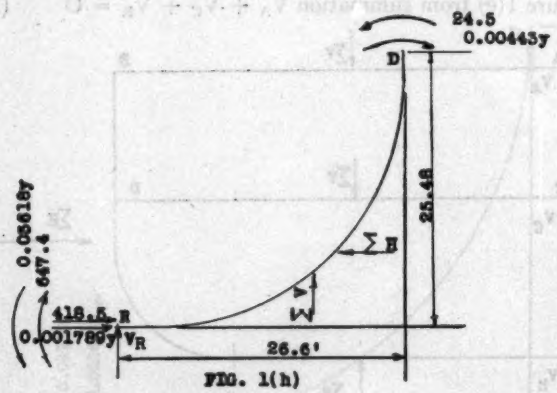
$$V_A = \frac{696.6 + 0.00043y}{26.6} = 26.19 + 0.0000162y \qquad (3a)$$

Similarly, in Figure 1(g) summation M about D = O,



$$V_C = \frac{535.2 + 0.00589y}{26.6} = 20.12 + 0.0002214y \qquad (3b)$$

In Figure 1(h) summation M about D = O,



$$26.6V_R - 418.5\,(25.48) - 0.001789y\,(25.48) - 0.03618y - 24.5 + 0.00443y + 647.4 + = O,$$

$$V_R = \frac{-4,239.5 + 0.07733y}{26.6} = -159.4 + 0.002907y \qquad (3c)$$

Substituting Eqs. 3 in Eq. 2 the value of y may be found; thus, $-113.1 + 0.003144y = O$, and $y = 35,970$

Step 5. The numerical values of the moments caused by the vertical displacement are determined by substituting the value of y in the moment terms as shown in parentheses in Figure 1(c). By summation of the moments thus obtained and those shown in parentheses in Figure 1(b), the following moments in foot-kips[5] are computed:

$M_{AB} = -118.8 - (0.00024)(35{,}970) = -127.4$; $M_{BA} = 69.1$; $M_{BD} = -69.1$; $M_{CD} = -179.3$; $M_{DB} = -40.8$; $M_{DC} = -50.9$; $M_{DR} = 134.8$; $M_{RD} = -654.0$. The horizontal thrust at R = 482.9 kips; and the vertical reaction = −54.8 kips. (The total compressive stress in stanchion RC = 2 x 54.8.) The vertical reaction $V_A = 26.8$ kips and $V_C = 28.1$ kips. By solving the same example by least work: $M_{RD} = -655.5$; the horizontal thrust at R = 482.3 kips and the vertical reaction at R = −55.2 kips. By comparison it can be seen that the two methods yield similar results.

Conclusion.

The method developed in this paper is general and can be used regardless of the number of decks in the frame. The typical example has only two decks but the additional work when there are more is small when compared to the solution of the many simultaneous equations required by the least work method. If an additional stanchion had been required another joint displacement, y′, would have to be solved. The typical example assumed no camber in the main or second deck; however, with additional work it can be included in the analysis.

References.

1. "Continuous Frames of Reinforced Concrete", by Hardy Cross and N. D. Morgan, John Wiley & Sons, Inc., New York, 1932, pp. 46 to 76.

2. "Analysis of Continuous Frames by Distributing Fixed-End Moments", by Hardy Cross, Transactions, Am. Soc. C. E., Vol. 96 (1932), pp. 1-156. Also see "Statically Indeterminate Structures," by Lawrence C. Maugh, John Wiley & Sons, Inc., New York, 1946, pp. 230-242.

3. "Analysis of Rigid Frames", by A. Amirikian, Government Printing Office, Washington, D. C., 1942, p. 390.

4. "The Column Analogy", by Hardy Cross, Bulletin No. 215, Engineering Experiment Station, Published by the University of Illinois, Urbana, Illinois, or refer to pp. 46 to 76 reference 4.

5. One foot-kip equals 1000 foot pounds.

6. "Analysis of Rigid Frames by Superposition," by David M. Wilson, Transactions, Am. Soc. C. E. Vol. 110 (1945) p. 1489.

FUNDAMENTAL RESEARCH FOR THE NAVY IN CERAMICS.

By R. D. Jackel,

Physical Chemist, Office of Naval Research.

Background.

The Author.

Roland Jackel, physical chemist with the Office of Naval Research, received his bachelor's degree in chemistry from the University of Chicago. After employment in the research laboratories of the Kellog Switchboard Company, Chicago, and the U.S. Rubber, Company, Mishawaka, Indiana, he entered the Naval Reserve and served from 1943-1946 aboard the anti-aircraft cruiser U.S.S. *San Juan* as an officer in the engineering department. Early in 1946 Lt. Jackel was transferred to the Office of Naval Research where he has since been in charge of the ceramics research program of that office.

During the last two years, the Office of Naval Research has activated an integrated, balanced program in ceramic research. Ceramics is that specialization of chemistry and physics which deals, historically, with the making of useful bodies from baked or fired clay. The term is derived from the Greek word *keramikos*, meaning potters clay; earthern ware. With the advent of modern technology and its many diversities, ceramics has taken on a much broader connotation embracing separate but well defined technologies such as refractories, glass, abrasives; and a series of somewhat isolated special fields such as electrical insulating and conducting ceramics, piezoelectric ceramics, ceramic fibers. This definition of the modern field of ceramics, if accepted, is not without some startling contradictions. For instance, diamond, the hardest of abrasives, is an allotropic form of carbon, and therefore can be claimed to be included in the field of organic (carbon) chemistry. Similarly graphite, one of the most refractory substances known, is also an allotrope of carbon.

Recourse to the periodic table does not give more precise definition to the chemistry of ceramics; the only generalization

that may be drawn is that all silicates of the metals are considered ceramic substances. Although the converse of the above generalization is not strictly true, silicate chemistry is generally regarded as the chemistry of ceramics.

Thus the field of ceramics is outlined largely along the lines of technological development and historical precedents with admittedly diffuse boundaries. The principles of this science are those of modern chemistry and physics; it is doubtful if any major fundamental modifications of the present concepts of matter and energy will come from research in ceramics, but neither will the great general principles of quantum physics imply sufficiently detailed instruction for the successful making of a ceramic body. It is then in the application of the methods of modern chemistry and physics to the quantitative elucidation and understanding of the behavior of matter and energy involved in ceramic technologies that the ceramics program of the Office of Naval Research is concentrated.

The sponsored ceramic research of the Office of Naval Research falls into four general categories: mechanical properties of ceramic substances, refractories, synthesis of minerals, and glass science. Representative projects in these fields are discussed in greater detail below.

Mechanical Properties of Ceramic Substances.

The study of the mechanical properties of ceramic bodies presents several unique features. Firstly, ceramic bodies are, as a rule, completely elastic; and hence have rather low breaking strengths in tension and cross bending. In compression, as might be expected, many ceramic bodies have strengths of the order of 300,000 p.s.i. (Figure 1.) This characteristic, "brittleness," of ceramics has dissuaded many engineers from even considering ceramics in their designs for highly stressed components. However, within the last year a ceramic-bladed gas turbine has been assembled and run at high speeds for hours at a time[1].

While there are many difficulties which must be overcome, this feat shows that ceramics can compete mechanically with metals in the very critical role of turbine blading. Figure 2 shows cross section pieces of some experimental German turbine

[1] Hartwig, S. J., R. W. Sheffin, and R. J. Jones, Preliminary investigations of a gas turbine with sillimanite ceramic rotor blades. N. A. C. A. Technical Note No. 1399 (July 1947).

FIGURE 1.
Triumph of porcelain over steel. This illustration shows the penetration and rupture by a ½" diameter specimen of a commercial porcelain during test in compression. The porcelain failed at 332,000 p.s.i.
National Bureau of Standards photo.

blades which were made in 1944-45. They illustrate the complexity of blade shapes, and the hollow blades are especially instructive. The finished firing of hollow shapes, such as blades No. 4 and No. 5, to a reasonable tolerance requires a high degree of laboratory and plant control.

The understanding of the mechanical properties of ceramics of bodies is the object of work being sponsored by the Office of Naval Research under the direction of Henry Eyring at the University of Utah. The concepts of Eyring's absolute reaction rate theory will be applied to ceramic systems. Instrumentation for the preparation and study of ceramic bodies is not yet com-

FIGURE 2.

Cross section pieces of experimental German turbine blades. On the basis of their composition, it is extremely unlikely that these blades would have been satisfactory for high temperature service.

No. 1—Alpha alumina dense, similar to "Sinter Korund".
No. 2—High glass body with quartz as crystalline phase.
No. 3—Similar to No. 2.
No. 4—Similar to No. 2 and No. 3. This shape was extruded.
No. 5—"Mesoenstatite" $MgO\ .SiO_2$. This shape was extruded.
No. 6—Similar to No. 5.

National Bureau of Standards photo.

plete. Data presently available in the literature on the creep, flow, and fracture of glass are being theoretically evaluated.[2]

Approximately one third of the extensive project at the University of Utah is devoted to the study of ceramic materials; the remaining phases are concerned with metals and organic fibres.

Interesting work on the mechanical property, abrasion hardness of single crystals, is being sponsored by the Office of Naval Research at the University of Michigan under the direction of C. B. Slawson. This project is a continuation of Prof. Slawson's work on the abrasion hardness of diamonds, and seeks to explain in a quantitative way how a crystal can be ground or abraded away with smaller particles of the same substance. If hardness were assumed to be a universal homogenous property of a single crystal, it would seem that a crystal could not be abraded with its own dust. The striking experimental contradiction to this hypothesis is, of course, the diamond, which can be faceted and polished only by grinding with diamond dust. An interesting outcome of Slawson's previous work is that diamond can be smoothly abraded only along certain directions, and while this fact had been known to diamond cutters for centuries, it has never previously been the object of a scientific investigation. In the present work at the University of Michigan similar data are being obtained on the vector abrasion hardness of hard single crystals such as periclase (MgO), spinel ($MgO.Al_2O_3$), and silicon carbide; (SiC).

This study is believed to be of greatest fundamental importance to the field of abrasives.

Refractories.

In the field of refractories there are at present four projects sponsored by the Office of Naval Research:

At Rutgers University a study of slagged boiler refractories is being made to elucidate the physico-chemical processes taking place in a brick during the firing of a boiler. A brick is, from the scientific point of view, an extremely complicated porous structure comprising numerous heterogeneous phases, mixed crystalline and glassy states, and in chemical composition only known to a good degree of approximation. The work in progress

[2] Baker, T. C. and F. W. Preston. Fatigue of glass under static loads. *Applied Physics*, *17*, 3, 170-178 (March 1946)

at Rutgers has thus far made some important progress in the quantitative study of naval boiler refractories: Formation of mullite ($3Al_2O_3.2SiO_2$) during the life of the brick at boiler temperatures has been traced by x-ray diffraction studies. Glass formation has been studied and several crystalline phases have been identified. Typical slags from naval boilers have been studied and correlation of the composition with the fuel oil used in firing the boiler has been attempted. The progress thus far has given clearer definition to the many problems in the physical chemistry of boiler refractories at operating temperatures. Results of this study are, of course, immediately applicable to the engineering problems involved in the operation and maintenance of oil-fired, steam-propelled naval vessels.

At the National Bureau of Standards work is continuing on special oxide refractories, or refractory "porcelain". Outstanding work in this field has been recently summarized and an excellent description of techniques and instrumentation for investigations of this kind can be found in the referenced work.[8] The best picture of this work at the Bureau can perhaps be given by briefly reviewing the various problems which are now under investigation:

(1) The zircon-silica system has been investigated, and one eutectic has been located at 1680° C. containing 10% zircon and 90% silica. Further studies are in progress to obtain the equilibrium diagram for this system.

(2) The system MgO-BeO-ZrO_2 has been extensively investigated. A ternary eutectic is believed to be located at the mole composition 3MgO: 5BeO: ZrO_2, fusing at 1690° C. Further detailed studies have been made in this system.

(3) Studies in the system $ThO_2.SiO_2$ are in progress. Thorite ($ThO_2.SiO_2$) has been synthesized and identified. This is the first reported synthetic preparation of thorite.

(4) Thermal shock and bending strength tests on the following bodies are in progress:

48 BeO	1 Al_2O_3	1 ZrO_2	– MgO
90 BeO	3 MgO	ZrO_2	
40 BeO	3 MgO	ZrO_2	
24 BeO	3 MgO	ZrO_2	
10 BeO	MgO	ZrO_2	
5 BeO	Mg	ZrO_2	

[8] National Bureau of Standards Research Paper, 1703 (1945).

(5) The system, MgO-BeO-ThO_2, is being investigated.

(6) (a) Further investigations in the fabrication of ZrO_2 furnace parts have been reported, essentially suppression of ZrO_2 (monoclinic) to ZrO_2 (tetragonal) transformation by addition of CaO. Figure 3 shows a zirconia furnace muffle which had failed

FIGURE 3.

Failure due to solid state reaction. This photograph illustrates a zirconia furnace muffle exhibiting the type of failure which was common before progress had been made on the stabilization of zirconia.

National Bureau of Standards photo.

after a few heating cycles because of the volume changes caused by the reaction ZrO_2 (monoclinic) ZrO_2 (tetragonal). Stabilization of zirconia by the addition of small quantities of other oxides has vastly increased the usefulness of this excellent refractory material.

(b) Further investigations have also been obtained in the fabrication of ThO_2-La_2O_3 resistor elements.

(7) Redetermination of melting points of refractory substances in the system Al_2O_3-MgO
56 Al_2O_3, 44 MgO (wt %)
98 Al_2O_3, 2 MgO (wt %) is being carried out. Also 2MgO-SiO_2 is being studied in some detail with respect to obtaining *pure* $2MgO.SiO_2$, and finding its melting point.

Support of the work on phase relations at the National Bureau of Standards is a major contribution to fundamental ceramic research in this country. While the work in itself may appear unspectacular, there can be no engineering application without these basic data.

At Alfred University, Alfred, New York, there is in progress under Office of Naval Research sponsorship a study on the phase relations of metals and ceramic substances. This investigation will establish the fundamental background for the possible techniques of fabricating refractory bodies of superior strength by using combinations of metals and ceramic substances. Preliminary empirical work done in the last few years indicates this approach to super-refractories to be entirely feasible and somewhat promising.

In this study small pellets of metal-ceramics formed by powder metallurgy techniques are quenched from furnace temperatures. X-ray diffraction patterns are then made to discover possible solid state reactions and determine structure. Petrographic examination and chemical analysis complement the x-ray work. Collateral studies are being made to determine the particle size distribution, atmosphere effect, and diffusion properties of the materials.

Thus far progress has been made on the following systems, working at 1400° C.: Ni-Al_2O_3, Ni-BeO, Cr-Al_2O_3, and Co-Be-O.

Investigations sponsored by the Office of Naval Research at Bettelle Memorial Institute are carrying out fundamental studies on bodies sintered from pure oxides. In this study attention is focused on bodies made from pure, sintered oxides, Al_2O_3 and BeO, and $ZrSiO_4$. Effects of (1) particle size and mode of preparation of raw materials, (2) mode of preparation of "green" body, and (3) firing schedule and refiring cycles are evaluated in terms of linear shrinkage in firing, water absorption, bulk density and

modulus of rupture values of the fired bodies. Petrographic examinations have been made on the fired bodies to determine crystal growth and glass formation.

Recent work has been concentrated on the growth of large grains in pure alumina bodies. In the firing schedules used, 3000° F. appears to be the threshold temperature for coarse crystal formation. Crystal growth progresses with time (at firing temperature) but it is not clear whether this is an exponential or linear process. Modulus-of-rupture results bear out the belief that coarse crystals reduce the mechanical strength of the bodies.

Interpretation of the results of this study on one component single-phase systems will be an important step in resolving many of the inconsistent beliefs about the general mechanical properties of ceramic bodies.

The groups of studies outlined in the foregoing will provide much of the fundamental ground work for the development of ceramics as engineering materials for use under extreme conditions of temperature and chemical reactivity. This fundamental work will also clarify the inherent limitations of ceramic materials and thus permit a rational approach to the development problems indicated.

The refractories projects have been discussed at some length because these are the projects which support and promote the exploration for new super refractory substances required for applications in rocketry, jet propulsion, and the transformation of nuclear energy for engineering application. Many of the devices now envisioned involve operating temperatures above the melting points of most metals, certainly all of the readily available metals, so it becomes necessary to turn to refractory ceramics for a possible solution to this problem.

Synthesis of Minerals.

Several interesting studies under Office of Naval Research sponsorship are in progress on the synthesis of minerals:

At the Electrotechnical Laboratory, Bureau of Mines, Norris, Tennessee, the synthesis of pure cordierite ($2Al_2O_3.2MgO.5SiO_2$) is being accomplished. The raw materials are being fused in a large arc furnace to produce cordierite glass, which is then devitrified to crystalline cordierite by appropriate heat treatment.

This mineral is of interest because it has an extremely low thermal expansion coefficient[4] over a large temperature range and excellent thermal shock resistance.

Phase studies are being made on the various compositions with which cordierite readily forms solid solutions[5], to define the optimum chemical composition of the starting materials for optimum cordierite formation in the melt. In addition to the chemical studies, accurate measurements will be made of the thermal conductivity, electrical conductivity, coefficients of thermal expansion, heat shock resistance and mechanical properties of pure cordierite bodies.

Also at the Electrotechnical Laboratory, the preparation of synthetic fluorine-phlogopite mica on a pilot plant scale is being undertaken. The mica crystals are to be grown directly from the melt.

This project is under the joint sponsorhip of the Signal Corps, Department of the Army, the Bureau of Ships, and the Office of Naval Research.

Production of synthetic mica on a commercial scale will make the United States independent of foreign sources for the strategic mineral that is widely used in Army and Navy communication and electrical equipment. Although natural mica is found in several parts of this country and despite a wartime government subsidy of $6.00 a pound, only 15 per cent of the amount needed in the critical war years of 1943 and 1944 was produced here. The Government has considered the mica supply sufficiently acute to designate needed grades of the mineral as Class A strategic materials under the recently enacted Stockpiling Act.

Known as fluorine-phlogopite mica, the synthetic form now being produced has the desirable characteristics of natural mica, including perfect cleavage into thin sheets, good electrical and mechanical properties, and chemical stability. It is expected to replace the muscovite and phlogopite forms of natural mica, silicate minerals which the United States has imported in large quantities for use as insulation in electrical machinery and as a dielectric in electronic circuits.

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[4] Reported to be 2.0 x 10^{-6} over the range of 20°C to 1200° C.

[5] Cordierite-mullite ($3Al_2O_3$, $2SiO_2$)
Cordierite-tridymite (SiO_2)
Cordierite-clinoenstatite ($MgO.SiO_2$)
Cordierite-forsterite ($2MgO.SiO_2$)
Cordierite-spinel ($Al_2O_3.MgO$)

Also inherent in the new mica synthesis process is the possibility that further research may reveal ways of directly fabricating mica components, which would eliminate the time consuming and laborious tasks of sorting, grading, splitting, and trimming natural mica.

It is interesting to note that although patents on mica substitutes have been granted from time to time during the last 30 years, no one had claimed the preparation of crystals of mica large enough to be split, like those found in nature. When technical intelligence teams operating in Germany in 1945 found that substantial progress toward the preparation of large mica crystals from a melt had been made at the Kaiser-Wilhelm Institute for Silicate Research, and that similar encouraging results had been obtained on larger scale by the Siemens-Halske Concern, Berlin, interest in the possibilities of synthetic mica was again stimulated.

The present Army-Navy research project is expected to solve the engineering problems involved in the synthesis of mica on an industrial scale, and to define the conditions under which the manufacture of mica may be economically feasible. It also will give the electrical and communication industries an opportunity to evaluate more fully the use of fluorine-phlogopite mica as an electrical insulating material.

At the National Bureau of Standards there is in progress a general study on the synthesis of fluorine-containing minerals and fluorine homologues of hydroxyl (OH^-) containing minerals. This research will generalize the conditions under which fluorine may be substituted for hydroxyl in various types of mineral lattices. The recent successful synthesis of fluorine phlogopite mica emphasizes the possibilities of obtaining easily crystallized silicates by the introduction of fluorine into the compound.

Hydrothermal bomb techniques will be exploited, as well as the more conventional means of preparation such as crystallization from a melt.

GLASS.

The Office of Naval Research has recently entered the field of glass science and two projects are now underway at Pennsylvania State College.

The first project is a study of the role of asymmetric groups in glass. This is a very fundamental investigation into the physico-chemical properties in glass of ions and oxide groups with certain electronic configurations. Oxide groups with low atomic coordination symmetry are known to have pronounced catalytic activity in many chemical reactions; these properties are thought to be caused by the distorted electron clouds of the asymmetric groups. In this research role of asymmetric groups in glass will be investigated in detail. Glass compositions, containing Pb^{++}, Bi^{+++}, and other ions which are known to form asymmetric oxide groups, will be examined for anisotropy of such properties as light transmission and dielectric constants. Such compositions will also be investigated for piezo-electricity. The effects of applying strong electrostatic and magnetic fields to glass containing asymmetric groups will be investigated, and the physical behavior will be correlated with the electronic configurations of the various oxide groups known to be present in the glass.

The other phase of the work on glass at Pennsylvania State College will be concerned with the effects of adsorbed ions on glass surfaces. It is known that adsorbed metallic ions play important parts in many technological processes involving glass. For instance, it has been shown that the proper adherence of metallic films (mirrors) cannot be achieved unless the glass surface is prepared by treatment with a solution containing certain metallic ions. In this research properties of various adsorbed ions on different kinds of glass surfaces will be explored further. Ingenious techniques such as studying the gas nucleation characteristics of a glass surface will be used. As a logical extension of this work, the effect of adsorbed metallic ions on the surface properties of ceramic glazes will be studied.

To further the understanding of the surface effects involved, the experimental results will be interpreted in terms of the symmetry of the electron cloud of the adsorbed metallic ion.

In guiding the growth of the ceramics program, close coordination with the broader fields of physical science as represented in the Physics, Chemistry and Geophysics programs of the Office of Naval Research has been of great value.

Conclusion.

Thus are reviewed the major fields of activity under the ceramics program of the Office of Naval Research. The list of projects is not complete; but rather those projects, which serve to exemplify the philosophy and plan of the program, were chosen for discussion.

DECAY IN BALSA LIFE FLOATS AND THE VALUE OF BRUSH AND DIP TREATMENTS WITH CHLORINATED PHENOL FOR ITS PREVENTION.

By Arthur S. Rhoads.

Introduction.

It is generally known that seasoned balsa (*Ochroma lagopus* Sw.) absorbs water readily but that this property can be overcome largely by the use of water-repellents or waterproof coverings which do not add greatly to its weight. The wood, by reason of its low specific gravity, high absorptiveness and lack of durability, is very susceptible to decay. It is regarded as being very readily attacked by fungi causing white rot but moderately resistant to those producing brown rot, and should be protected against moisture in service (4).

The Author.

The author is a forestry graduate of The Pennsylvania State College, with an M.S. degree from the same institution and a Ph.D. degree from The New York State College of Forestry at Syracuse University. Since completing his post graduate work he has devoted 31 years to research on wood decays and wood-rotting fungi, and diseases of deciduous fruit trees, small fruits and citrus, serving mainly with the U. S. Department of Agriculture and State Experiment Stations. He is at present engaged in studies on marine borers with the William F. Clapp Laboratories, Inc., Duxbury, Mass.

In the commercial manufacture of balsa life floats up to the present time no effort has been made to apply either any wood preservative or water-repellent, aside from the waterproofing compound applied to the surface in the wrapping procedure. Neither is it customary to apply any preservative to the canvas used in wrapping. It has been estimated by the Bureau of Ships that an annual replacement of 50% of the life floats in use is necessary. While this estimate includes floats lost at sea and those damaged in various ways, the percentage retired from service because of decay of the wood following deterioration or injury of the covering is undoubtedly large. It is obvious that

the application of an effective preservative treatment to both the wood and the canvas covering in the manufacture or repair of life floats would do much to eliminate this unnecessary and unwarrantable early deterioration, and materially increase their service-life.

In July, 1946, the Bureau of Ships authorized the Industrial Test Laboratory of the Philadelphia Naval Shipyard to determine the effect of wood preservatives on the durability of life floats, as well as to investigate a number of other factors in connection with their construction and testing. The investigations here reported had their inception in the development of a quick and effective procedure for reconditioning old life floats on hand at this Shipyard (5). That report recommended as part of the procedure, that a heavy brush coat of chlorinated phenol (Type B) preservative solution, conforming to the Bureau of Ships specifications (3), be applied to the bare wood bodies of the floats after stripping off the old canvas covering and sanding. The brush treatment was used because no facilities were available for dipping and it was not deemed expedient to have a vat constructed for the relatively small number of floats that were being treated. It was apparent from the deteriorated condition of many of these and other floats examined that there also was need for study of the possibility of using an effective wood preservative during new float construction. This study had three objectives:

(1) To make a survey of the decay occurring in balsa life floats and determine the rôle of moisture absorption in favoring its development.

(2) To work out a quick, economical and effective means of treating the wood with a preservative solution which would increase its resistance to decay without unduly decreasing its buoyancy or otherwise adversely affecting its other properties.

(3) To determine the effectiveness of such a preservative treatment in preventing decay of balsa by actual test against wood-rotting fungi of wood so treated.

Sources of Balsa Life Floats and Lumber Used in the Investigations.

The balsa life floats studied during the course of these investigations were principally those in outdoor storage at the Shipyard. The stock consisted of floats which had been discarded by ships

and were earmarked for ultimate repair, shipment to other agencies or scrapping, depending upon their condition. Floats of several different types and sizes were represented. The storage area is low and so poorly drained that water stands at the lower end after rainy periods. No stickers were used, though occasionally small pieces of 2 × 4's were inserted between the floats. As a rule the floats were separated only by the canvas straps or occasionally by rope from the lashings, the latter resulting in denting due to the softness of the wood and the weight in stacking floats to as many as 18 deep. The closeness of the stringers to the ground and of the floats in the stacks, together with the lack of spacing between the stacks, as shown in Figure 1, greatly retarded air circulation and drying after rainfall.

A few 25-person life floats from a group of 12 which were manufactured at the Shipyard in connection with a test of various wrapping procedures using an approved coating (No. 824 Balsa

FIGURE 1.

Life floats of various types in outdoor storage at this Shipyard, showing stacking close to the ground in a low, poorly drained situation, with lack of stickers between floats and closeness of stacks which retards circulation of air and drying after rainfall.

Wood Coating, manufactured by the Akron Paint and Varnish Company, Akron, Ohio) also were examined. A detailed description of the construction and covering of these floats has been given elsewhere (7). They had been subjected to the usual submergence test under 10 feet of water for 24 hours as required by the specification (8) and had been stored behind the Laboratory, exposed to the weather, since about November, 1944. During the submergence test 7 of these floats, as reported previously (7), absorbed an average of 56 pounds of water, an amount materially exceeding the specification limit of 40 pounds.

All balsa lumber used in these investigations came originally from the stock at the Shipyard. Some was secured in the form of short lengths remaining at the Joiner's Shop after the construction of life float sections for experimental purposes. Short sections remaining from previous tests at the Laboratory also were available.

Occurrence of Fungi and Decay in Balsa Life Floats and Lumber.

Unfortunately, opportunities for studying the occurrence of fungi and decay in balsa life floats and lumber at this Shipyard have been limited. Aside from the 50 floats and a few experimental floats which were stripped in the course of development of the reconditioning procedure, no close inspection of the wood bodies of stored floats was possible and moisture readings could not be taken. Similarly, there was only the most meager opportunity to study the occurrence of decay in balsa lumber since it is only rarely that the present needs of the Shipyard call for dressing and cutting any of the small stock supply remaining.

Early in the summer of 1946 the cosmopolitan xerophilous fungus, *Schizophyllum commune* Fr., was found to have started fruiting on a partially stripped float from the experimental lot of 12 stored behind the Laboratory. These floats had absorbed large quantities of water in the submergence test so that the moisture content of the wood was unusually high and outdoor storage, with exposure to rainfall, caused it to remain so. By midsummer, this fungus was fruiting more or less abundantly on 4 of these floats and new fruiting bodies developed successively as the old ones deteriorated. New crops of these developed abun-

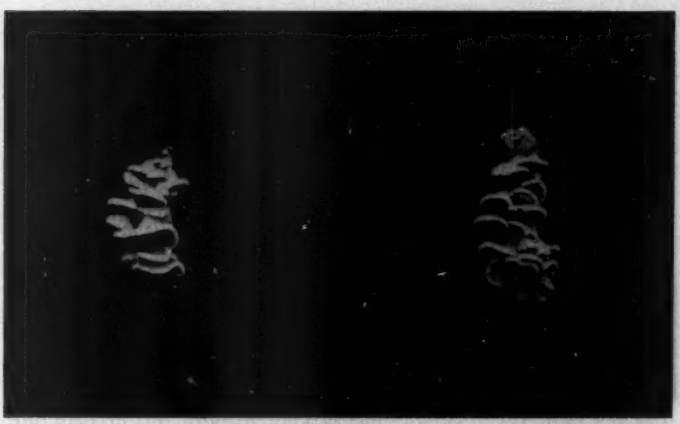

FIGURE 2.
Close-up view of balsa life float, showing fruiting bodies of *Schizophyllum commune* developing at laps in canvas covering.

dantly on the parts of the floats least exposed to the sun and drying out of the wood, particularly along the lower margin of one float resting on top of another and on the end of one close to the building (Figures 2 and 3). The fruiting bodies developed most readily from the exposed laps in the canvas covering and from areas where this had been stripped (Figure 4). However, in some cases they also grew directly through the unbroken covering. Following the taking of moisture readings on the floats, a striking tendency was observed for fruiting bodies to begin developing at one or more of the series of 4 punctures made through the covering by the contactors of the mositure meter. It is thus apparent that the intact canvas covering exercised a marked inhibiting effect on the development of fruiting bodies, though not on the development of decay. As a matter of fact, after water has once gained entrance, the covering retards the drying out of the wood body and creates a moist-chamber effect, which expedites the vegetative growth of the fungus and the development of decay. The abundant development of this fungus through various punctures in the cover resulting from the taking of moisture readings, or at places where the cover had been partially stripped, suggests that the wood was infected prior to fabrication into floats.

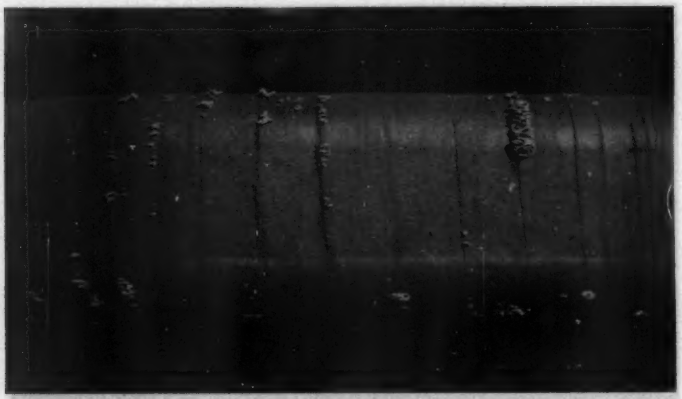

FIGURE 3.
Later photograph of balsa life float illustrated in Figure 2, showing abundant fruiting of *Schizophyllum commune* mostly at laps in canvas covering, but also growing directly through it.

FIGURE 4.
Balsa life float showing *Schizophyllum commune* beginning to fruit on wood exposed where canvas covering was stripped.

On one of the stripped floats on which *Schizophyllum commune* fruited to some extent a few tan-colored masses, suggestive of the conidial stage of a Pyrenomycete, were found growing from the transverse end where a length had been cut out for making cultures. A section of this float bearing these immature fruiting bodies was sawed off and placed in the humidity control room in order to stimulate their development, after first making tissue cultures. Unfortunately, however, roaches infesting this room at the time ate the young fruiting bodies down to the wood by the following morning. No further development of this fungus occurred on this float section, though *S. commune* fruited for a while, and later a slime mold (*Fuligo septica* (L) Weber). Fortunately, however, a further development of the conidial stage of the Pyrenomycete occured later from both the newly cut surface and also on the opposite end of the remains of the float left outdoors, and this later formed the perfect stage and the fungus was identified as *Daldinia concentrica* (Bolt.) Ces. & De Not. (Figure 5).

The life floats stacked in the outdoor storage area were inspected in July, 1946, again later in the summer, once in the fall,

FIGURE 5.
Balsa life float showing *Daldinia concentrica* fruiting on wood exposed by stripping canvas covering, with traces of conidial stage still showing.

and finally in June, 1947. These could be examined only superficially for evidence of decay at points where breaks occurred in the lacquered canvas covering, and for the occurrence of fruiting bodies of wood-rotting fungi. In the late summer and fall of 1946 *S. commune* was found fruiting on 7 obsolete, oval-type floats, and on 2 rectangular floats which had rotted areas of wood broken out and obviously were worthless. A considerable number of floats, especially where the covering had been damaged showed evidence of decay though no fungus was fruiting. On June 9, 1947 the greatly reduced stock in the outdoor storage area (mostly a different lot than those examined previously) was again examined. Wood-rotting fungi were found fruiting on two floats of the rectangular type—*S. commune* on one, and *Polyporus adustus* Willd. ex Fr. on another which had been stripped of its covering.

Fifty of the better appearing floats of the rectangular type selected earlier from the outdoor storage area as probably suitable for use in working out a quick reconditioning procedure (5) were examined in detail after stripping the canvas covering and sanding. These, of course, did not represent the worst conditions to be found in this lot of floats. Several of those selected showed some deterioration of both the canvas covering and the underlying wood, especially around damaged places, and molds, as well as an Actinomycete, were found growing under the canvas in places where deterioration or injury of the covering had led to high water absorption and development of decay. Three of these floats were discarded because of decayed wood which would have required replacement, and two others were discarded on account of open glue joints. Attempts were made to isolate the wood-rotting fungi involved by means of an extensive series of cultures made from bits of wood removed from the interior of these floats, using chisel-forceps with the usual aseptic precautions. These attempts were unsuccessful, however, as the cultures in all cases were quickly overrun by molds which developed from the bits of wood used as inoculum. It was found that the various glued pieces of wood comprising the floats invariably were thoroughly permeated by mold fungi, even in cases where the wood appeared sound. In fact, wood cut from floats frequently was so thoroughly permeated by such fungi that it had a strong moldy odor.

FIGURE 6.

Longitudinal and transverse views of balsa wood cut from life float, showing blackish zone lines denoting incipient decay present in wood when float was manufactured.

Pieces of wood in a number of these floats showed prominent dark sepia-brown zone lines indicative of incipient decay, yet with no appreciable softening of the wood (Figure 6). Similar zone lines also were observed in pieces of wood used in gluing up some short float sections which had been made by the Joiner's Shop for the experimental testing of covering methods. This shop was visited later on an occasion when balsa lumber was being dressed preparatory to making up an additional supply of life float sections, in order to observe the decays and defects occurring in the wood. Some timbers were found to be largely involved in incipient decay and dark brown zone lines. In cross sections the lighter-colored areas of incipient decay were demarked from the sound wood by a highly irregular development of zone lines, fluctuating radially along the periphery, with occasional small, radially-elongated islands bounded by zone lines in places. In longitudinal sections the wood of the affected areas, which often extended for several feet, was characterized by greatly elongated areas bounded by zone lines, local islands or streaks which gave the wood a somewhat mottled appearance, or by combinations of these (Figure 7). The areas included within these zone lines, whether large or small, were characteristically wider in the radial than in the tangential plane but of far greater extent in the longitudinal direction, and often aggregated into areas extending several feet longitudinally. Examination of stained microtome sections of these zone lines showed that in both the old life floats and the new lumber they were caused by localized dense concentrations of dark brown mycelium within narrow zones, ranging from 25-75μ broad, extending irregularly through the wood. The mycelium often was characterized by globular to elliptical enlargements. In some cases isolated vessels were completely occluded. In fine blackish zone lines in part of a life float with a whitish decay of the wood, the dark brown mycelium in the zone lines was characterized by densely massed, globular cells 7.5-18.5μ in diameter. While such incipient decay undoubtedly occurred in the wood prior to procurement, the fungus appears to have been killed in kiln-drying and showed no evidence of having made any further developemt in the wood used in life floats.

Other balsa timbers in the shop exhibited circular areas of brittle brown wood surrounding the pith, which in some cases had

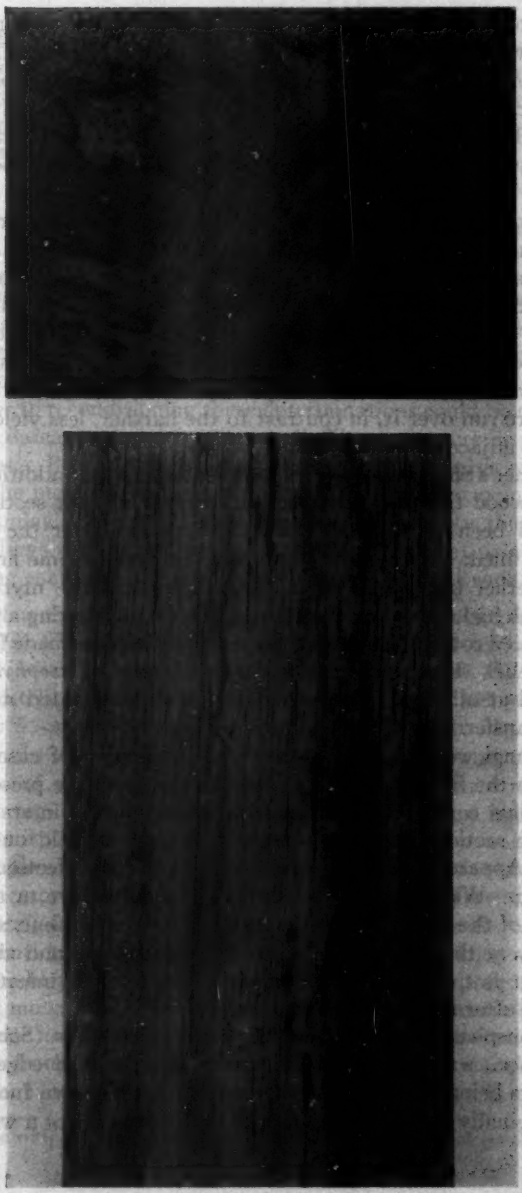

FIGURE 7.

Longitudinal and transverse views of balsa wood from shipyard stock, showing blackish zone lines denoting incipient decay apparently present when wood was delivered.

disintegrated. This wood was characterized by extensive honeycombing, which obviously developed during kiln-drying. This decayed zone was distinctly conical in vertical extent and appears to be the condition known as "water-heart" (17). Some of the extremely light-weight pieces showed an abundance of fine transverse fractures extending rather deeply in the outer part. Some timbers, when dressed, showed much of the wood involved in a pale, straw-colored rot contrasting rather vividly with the normal wood. Such decayed wood had a softer and more velvety feel than the adjacent sound wood and often was so soft that it would sink as the ends of the fingers were run over it with a light pressure. The softest longitudinal streaks were characterized by minute pockets extending longitudinally. On cross sections the pale straw-colored areas felt like the pile of velvet when the fingers were run over it, in contrast to the harsher, less yielding feel of the adjacent sound wood.

The Joiner's Shop was visited subsequently to secure additional pieces of wood from lengths too short for use in float sections, which had been stacked under the inside stairs after the last order was filled. Two pieces were found to have become firmly stuck together by the development of a small white mycelial sheet which had grown out from a piece of plank showing a pale straw-colored rot similar to that described. Cultures made from this mycelial sheet yielded what was clearly *Schizophyllum commune* but all were overgrown later by an associated mold, despite transferring twice.

Mold fungi were observed to occur in a number of cases in balsa from the Shipyard's stock and in some cases the presence of such fungi could be demonstrated microscopically in stained microtome sections of wood with no evidence of mold on the surface. Apparently balsa is very susceptible to infection by such fungi. Whether these infections occurred before or after the entry of the wood into this country is not known but, since the supply at the Shipyard is said to have been on hand about 3 years, it is quite likely that most of these mold infections developed after receipt of the lumber.

An inconspicuous Ascomycete, *Mycogala parietinum* (Schrad. ex Fr.) Sacc., was found fruiting on the rough-sawed edges of two planks being milled in the shop. This little known fungus, which originally was described as a Myxomycete, is not a wood-

rotter, however. Its identification was made by Miss Edith K. Cash of the Division of Mycology and Disease Survey, U. S. Department of Agriculture. The Mycological Collections of this office contained but two previous specimens of this fungus—one on mushroom compost and the other on filter paper.

Moisture Determinations on Balsa Life Floats in Outdoor Storage.

Moisture determinations were made with a Kaydel moisture gauge (Type L-4102) on 39 of the 50 life floats selected from the outdoor storage area for use in working out a quick reconditioning procedure. Since the covers were to be stripped from all of these prior to reconditioning these determinations could be made without damage caused by the needle-like contactors of the instrument. Preliminary readings showed a high degree of variation in the moisture content of the wood at different points prior to stripping. In order to standardize and simplify the procedure readings were taken on the inside and outside at points midway along each side and end, making a total of 8 for each float.

Of a total of 312 readings taken on 39 floats, only 119, or 38.1%, were below the fiber saturation point (24-25%), the readings indicating an average moisture content of 47.7%.[1] Only 10, or 25.6%, of the floats gave readings averaging below 25%, and only 5, or 12.8%, gave no readings above 24%. It was found that the condition of the floats could not be evaluated on the basis of the external appearance but only after stripping the covering. Decay was present in several showing a high moisture content, being confined largely to the lighter-weight wood. Moisture determinations, similarly made on 13 floats which had been stored indoors, showed a very low moisture content in general, though 9 showed one or more readings above the fiber saturation point. Unfortunately, nothing was known in regard to the history of any of these floats or as to how long the locally high moisture content so generally exhibited by those in outdoor storage had been maintained. Three significant points are quite, clear from the examination of the stored floats, namely (1) that the canvas covering frequently proves ineffective in preventing

[1] The writer is indebted to Mr. F. M. Willson for the date on moisture determinations.

the absorption of water and may deteriorate due to the activity of mold organisms; (2) that the high moisture content shown in places in many of the floats, if maintained over any considerable period, would prove highly conducive to the development of decay; and (3) that balsa is very susceptible to such decay, which frequently does occur under existing conditions.

Absorption of Chlorinated Phenol Solution by Balsa.

It is stated (5) that a certain amount of preservative can be injected into the vessels of balsa quite easily by either pressure or open tank processes but that other parts of the wood are difficult to impregnate. The wood is therefore classed as resistant to impregnation though the sapwood is considered permeable. It is further stated that balsa is liable to collapse and distort when treated with creosote under pressure. However, the present study was concerned only with the applicability and penetration of brush and dip treatments.

In order to determine whether or not balsa could be effectively treated by a simple and economical procedure a number of tests were made of its absorptive capacity for chlorinated phenol (Type B) preservative conforming to the Bureau of Ships specication (4).[1] This preservative was selected because of its availability and the fact that previous tests at this Laboratory (18: 19) had demonstrated that, of the three types covered by the specification, Type B proved the most effective in preventing decay in wood and in preventing the spread of decay from infected to sound wood. Preliminary tests were conducted on 1-inch square air-dried pieces in 10- and 20-inch lengths. Paired light- and heavy-weight pieces were used for each treatment, those 10 inches long having the ends sealed while the 20-inch pieces were left with the ends unsealed. Weights of the wood used ranged between 4.5 and 14.5 lbs. per cu. ft. on an air-dry basis. The sticks were weighed before and after dipping and draining, the difference representing the preservative solution absorbed. Four methods of treatment were tried: a 2-coat brush application, a 1-minute dip, a 30-minute dip, and a 5¾-hour immersion. The average absorption for the 1-minute dip was found to be approximately

[1] The specification stipulates that this shall consist of at least 159 gms. of chlorinated phenols per gallon of solution, of which at least 95 gms. shall be pentachlorophenol. The remaining 64 gms. may consist of chlororthophenylphenol, tetrachlorophenol or any mixture of them.

twice that for the 2-coat brush treatment, while for the 30-minute dip it was approximately one-half greater than for the 1-minute dip, and for the 5¾-hour immersion approximately 3 times greater than for the 1-minute dip, or over twice that for the 30-minute dip, based on the 20-inch pieces with the ends unsealed. In the 10-inch pieces with the ends sealed the average absorption in 5¾ hours was approximately 2½ times that for the 30-minute dip. From a comparison of the pieces with the ends sealed and unsealed, respectively, it was apparent that most of the preservative solution was absorbed through the ends of the vessels and that comparatively little was absorbed radially or tangentially except where the grain of the wood was diagonal. The ratio of lateral to longitudinal absorption was about 1 to 4 or 1 to 5 in the few specimens used. The absorption of the preservative solution was materially increased by the presence of wormholes, a common defect in balsa.

A more comprehensive test of the comparative absorption of the preservative solution by brush and 3-minute dip treatments was conducted on a number of 3½ × 5 × ¾-inch blocks which were subsequently used in decay tests. One hundred and forty of these, ranging in density from 4.8 to 22.3 lbs. per cu. ft. on an oven-dry basis, were arranged in series according to increasing weight, beginning with the lightest, and numbered consecutively. Every tenth block, beginning with No. 1, was given a brush treatment, and every tenth block, beginning with No. 10, was given a 3-minute dip. The remainder of the blocks were left untreated for decay tests. A total of 28 blocks were treated. These were weighed after oven-drying and allowed to cool before dipping. They were weighed again immediately after dipping and draining, the difference being recorded as the weight of the preservative solution absorbed.

These results showed considerable variation in the amount of the preservative solution absorbed in both treatments. In the 3-minute dip three of the blocks (one with wormholes) absorbed twice as much of the solution, and one about 2½ times as much as the general run of the others. The blocks absorbed an average of 7.1 gms. in the brush treatment and 11.2 gms. in the 3-minute dip. No correlation was evident between the density of the wood and the amount of preservative solution absorbed in either treatment, and the amounts absorbed in each varied within wide

limits. However, the absorptive capacity was sufficiently high to warrant the assumption that a dip treatment would impart adequate decay resistance to the wood for the purpose desired.

An additional test of the absorptive capacity of balsa for chlorinated phenol (Type B) preservative solution in a 3-minute dip was made on a series of 76 oven-dried blocks 1½ in. square by 4 in. long. They ranged in weight from 9.3 to 59.7 gms., or from approximately 4 to 25 lbs. as calculated on a cu. ft. basis. The average absorption of the preservative solution was 10.4 gms. and the average weight of the blocks prior to dipping was 25.1 gms., or 10.6 lbs. as calculated on a cu. ft. basis. It again was evident from the results that balsa is extremely variable and erratic in its absorption of this preservative solution. Blocks of the same size and more or less identical in weight, and in some cases even cut from the same piece of wood, often show amazing differences in the amount of solution absorbed. This is particularly true in light- and medium-weight wood. It was noted that blocks cut from a small piece of wood less than ¼ cu. ft. in volume ranged from 9.8 to 15.5 gms. in oven-dry weight and absorbed amounts of solution ranging from 7.2 to 25.4 gms. In some cases where there were two blocks of the same or nearly the same weight, one occasionally absorbed up to five times as much as the other.

A similar test was made on oven-dried blocks of various conifers and hardwoods of the same size to compare their absorptive capacity for chlorinated phenol solution with that of balsa in the 3-minute dip. The woods used comprised bald cypress, Douglas fir, western white pine, southern yellow pine, ponderosa pine, white oak (two lots), red oak (three lots), white ash, and sugar maple. In each case three adjoining blocks were cut from heartwood and the absorption of the chlorinated phenol solution was averaged. These blocks ranged in weight from 53.0 to 97.4 gms., or from 22.4 to 41.2 lbs. as calculated on a cu. ft. basis. The average absorption of chlorinated phenol solution ranged from as low as 0.9 and 1.5 gms., respectively, in the two lots of white oak to 4.8 gms. in ponderosa pine, and even up to 6.1 gms. in one lot of red oak, though in the latter case most of the solution was retained in the vessels. It thus appears that balsa, in general, is much more absorptive of the preservative solution than the other woods tested for comparison.

The course of the absorption of chlorinated phenol solution by balsa was followed readily by dipping other blocks in this solution to which an oil-soluble dye had been added. When but one end of a block was dipped into the solution it was seen to be absorbed readily by the vessels in the wood above the level of the liquid. When blocks 3 or 4 inches long were dipped the colored solution appeared at the top in first one vessel and then in another in but a few seconds. It is thus apparent that the solution rapidly penetrates the vessels, which act as capillaries to draw it up, and that it diffuses from the vessels into the adjacent medullary rays and masses of wood elements between the rays.

That the medullary rays aid materially in the absorption and radial diffusion of the preservative solution can be demonstrated readily by sealing the end surfaces with thick collodion before dipping them in the dyed solutions. In blocks thus treated absorption through the ends of the vessels is completely prevented. When such blocks were split open following a 3-minute dip there was found to be considerable absorption and radial diffusion of the colored solution through the medullary rays and thence into the cells between the rays and occasional vessels, thus reversing the course of absorption described in the preceding paragraph. A still greater diffusion occurs when the blocks are allowed to stand for a while. Radial diffusion through the ray parenchyma cells is not uniform but much stronger in some of the rays than in others.

Most of the blocks tested that weighed between 4 and 5 lbs. per cu. ft. absorbed amazing quantities of the preservative solution, with air often still bubbling out of the vessels on the ends at the conclusion of the 3-minute dip, thus indicating that their capacity was by no means reached. However, even for quite a while after dipping, the solution would emerge readily on the surface when indented with the thumb. Upon splitting such blocks, the solution appeared to have diffused throughout the entire block in each case. In blocks weighing from 5 to 15 lbs. per cu. ft. the absorption was much lower and fairly consistent in general, with occasional specimens proving exceptionally absorptive. Blocks weighing at the rate of from 15 to 25 lbs. per cu. ft. showed the lowest and most consistent rate of absorption and had the least evidence of penetration and diffusion when split open. In very heavy pieces the solution showed but scanty

penetration in the 3-minute dip and remained mostly on the surface. In the light-weight balsa, blocks of the same weight showing high and low absorption, respectively, proved so similar in appearance and structure that even microscopic examination of stained microtome sections failed to disclose any consistent differences which would account for the extreme variations in absorption. However, there was some indication that in blocks showing low absorption the pits in the walls of the ray parenchyma cells were smaller and less abundant.

Extreme Variability of Balsa.

The results of these investigations, particularly those obtained in tests of the absorptive capacity of balsa for chlorinated phenol solution, emphasize the extreme variability of this material. The variability of the wood as regards density and, to a lesser extent, strength, has been pointed out in a previous report (6). Some further discussion of the extreme variability commonly exhibited by this wood seems desirable, however.

Wood, being an organic product developed by trees growing under widely varying conditions, naturally is subject to considerable variation. Of the various woods, however, probably none shows greater variation than balsa. In fact, the only thing about this wood which appears consistent is its extreme variability. However, this is to be expected in a tree growing with such phenomenal rapidity that it may attain a height of 50 to 60 ft. and a trunk diameter of 24 to 30 inches in 5 or 6 years (14; 17). Another author (16) states that in 6 months from seed a balsa tree in Ecuador will be 25 ft. high and 5 inches in diameter, and in 3 to 5 years it will be over 2 ft. in diameter at the base. The wood produced by such rapidly growing trees naturally is very light and spongy, weighing only 6 to 8 lbs. per cu. ft., though timber of slow growth, whether from the outer part of old trees or from young trees which developed under adverse conditions, may weigh up to 25 lbs. per cu. ft. (14). In order to secure a product with anything approaching a reasonable degree of uniformity the trees must be grown in plantations.

The pieces of balsa which have been examined in the stock at this Shipyard have been found to show an even greater variation, as was pointed out previously (6). Densities have been observed

ranging from as light as 4 lbs. to as heavy as 25 lbs. per cu. ft. The Navy Department specification (12) requires that the wood shall weigh not less than 8½ nor more than 12 lbs. per cu. ft. when kiln-dried to a moisture content not exceeding 12 per cent. The weight of balsa often varies within wide limits in different pieces cut from the same log or bolt. In tests on a bolt about 17 in. in diameter and 5 ft. long shipped from British Honduras the following variations were noted in the weight per cu. ft. of the dry wood at different distances from the pith: Center to 3 in. radius, 7.5 lbs.; from 3 to 6 in. radius, 16.6 lbs.; and from a 6-in. radius to the periphery, 21.5 lbs. (1). Experience with balsa at this Shipyard, as reported previously (6), shows that the weight often varies within wide limits in different pieces cut from a single sawn plank or timber and that the range of variability increases, within certain limits, the more a piece is subdivided. For example, in one case where a small squared piece had a weight of about 14 lbs. per cu. ft. the blocks cut from it varied from around 10 to nearly 20 lbs. per cu. ft.

While balsa was in great demand during the war for many purposes, particularly for aircraft and life floats in which specifications called for wood within a certain weight range, the extremely wide range of weight variation in the wood as received from the supplier often proved an important factor limiting its use. As was pointed out by Limbach and Paul (11), the extreme variation in weight of balsa makes its classification into weight grades essential if a buyer is to obtain a shipment best suited to a given purpose.

Microscopic examination of differentially stained microtome sections of light-weight, medium-weight and heavy-weight balsa showed marked differences in structure which make it clear why the wood varies so much in its appearance and physical properties. In the light-weight wood the structure closely approximates that of a piece of pith in that virtually all the cells are extremely thin-walled, with none except the vessels and occasional, more or less scattered cells showing any evidence of secondary thickening or lignification. In such wood the medullary rays are of two types, uniseriate and multiseriate, and consist largely of empty cells of the same general type as tile cells but much larger. The pits to vessels, as well as from one ray cell to another or to wood parenchyma, are quite large. Such

wood is often pale straw-colored, though sometimes pale grayish-brown and with rather narrow, inconspicuous medullary rays.

In medium-weight wood a large proportion of the cells show more or less lignification. This does not always occur uniformly throughout the wood and may be more prominent in occasional zones and less so in adjacent parts of a cross section.

In heavy-weight wood the majority of the cells show considerable lignification and are smaller in general in cross section than those in the light-weight or medium-weight wood, and the multiseriate rays are much broader than those in lighter-weight wood. The rays are distinctly heterogeneous, showing an abundant development of parenchyma cells in addition to the thin-walled marginal tile-like cells. The latter contrast strongly with the procumbent cells of the ray parenchyma comprising the interior portions. As the wood increases in density it often develops a pale brown color and the medullary rays become darker and more nearly pale golden brown against a lighter background, and much more prominent. In heavy-weight wood the prominence of the medullary rays in radial and tangential sections gives it a close superficial resemblance to sycamore (*Platanus*). Since a pale brown color sometimes occurs in light-weight wood, the color of a piece is by no means always an indication of its weight but, as a rule, a brownish color indicates a dense wood.

Decay Test of Treated and Untreated Balsa in Block Assemblies Against Schizophyllum Commune.

Experiments were designed to test both the capacity of *Schizophyllum commune*, a common wood-rotting fungus found on a number of life floats in outdoor storage, to produce decay, and the toxic effect of chlorinated phenol preservative solution on the development of this fungus. The major experiment utilized the series of 140 balsa blocks (3½ × 5 × ¾-in.) previously described as having been numbered consecutively after arranging in order of increasing density, with every tenth block, beginning with No. 1, given a brush application, and every tenth block, beginning with No. 10, given a 3-minute dip. Each block of this series had an inoculum block of balsa of the same size attached to it by a heavy rubber band. These block assemblies were set up on edge on aluminum racks in a humidity control room.

The inoculum blocks used were prepared by inoculating with the test fungus in heavy galvanized cans in which well-soaked blocks were packed in soaked cracked corn. After placing a ¼-inch layer of corn in the bottom of the culture can, the blocks were packed on edge and corn added to fill all spaces between any blocks not in close contact with one another and between the outer blocks and the can, and to provide a layer over the top. A second layer of blocks and corn was added in the same way except that one block, immediately below a hole in the lid of the can, was omitted to provide a place for the introduction of the inoculum. A heavy strip of non-absorbent cotton was placed around the rim of the can and the cover was forced down over it, making a good seal, and a cotton plug was placed in the hole in the cover. The packed cans were sterilized in an autoclave at 15 lbs. pressure (240° F.) for 4 hours, and were resterilized after a lapse of one or two days, using the same pressure and time. After each can had cooled from the second sterilization, inoculum from pure cultures of the test fungus was introduced through the hole in the lid, placed near the center and forced down into the moist corn in the channel which had been left for this purpose. Long forceps flamed in alcohol were used to introduce the inoculum and the cotton plug was replaced immediately. The inoculated cans were incubated at about 85° F. for a period of 3 months. Upon opening them at the end of this time a fine growth of the fungus was found to have developed throughout, with pronounced decay apparent in many of the blocks. These inoculum blocks were removed and attached to the test blocks after scraping off the corn and excess mycelium.

The block assemblies were installed in the humidity control room on December 3-5, 1946 and the room was humidified nearly every day to maintain a high degree of moisture. The test fungus grew well for two months and readily invaded the untreated blocks but showed no evidence of growth on any of those given either the brush treatment or the 3-minute dip. In but a few days it began fruiting on many of the inoculum blocks and also fruited later on many of the untreated blocks placed in contact with them. Unfortunately, the full measure of success which should have been possible in this test was not realized since there was no means of controlling the temperature within the humidity control room and, during the cold weather, steam

pipes passing through the room frequently raised its temperature overnight to as high as 94° F. While the assemblies were maintained until March 25, 1947 before removal for final readings, the fungus apparently ceased growing after the first week in February and gradually began dying out, after which a *Chaetomium* mold frequently began growing on a number of the assemblies. However, it was observed that the treated blocks, including even those given the brush treatment, remained clean and bright and free from mold.

Upon removal from the humidity control room the inoculum blocks were discarded and the test blocks were brushed clean of mycelium and oven-dried until weight loss ceased. The extent of decay in these blocks was judged by the loss in weight during the exposure to decay.

The figures on weight losses were averaged for the 8 untreated blocks in each group of 10 arranged in sequence according to density. The average weight losses in these fourteen groups of blocks ranged from 7.5 to 21.4%, and averaged 13.9%, being greatest in the light-weight, and least in the heavy-weight blocks.

The weight losses for the 14 blocks given the brush treatment and 3-minute dip, respectively, were calculated on the basis of a 15% volatile content of the treating solution since these blocks, after treatment, could not be oven-dried without loss of the preservative. In both methods of treatment the calculated weight losses were very small and, because of the method of calculation, of little significance. In both the brush and dip treatments the weight of the blocks at the conclusion of the test was greater than it was prior to treatment. It is thus evident that even the brush treatment left sufficient toxic residue on the wood to prevent attack by *Schizophyllum commune* during the period when the untreated blocks invariably were more or less decayed.

Decay Tests of Treated and Untreated Balsa Blocks Against Poria Monticola[1] and Lenzites Trabea.

The only additional decay test which the writer was able to make before the termination of this work was a series of 12 jar cultures designed to test the effectiveness of *Poria monticola* and *Lenzites trabea* in the decay of balsa blocks and, at the same time,

[1] Formerly designated as *Poria microspora*.

FIGURE 8.

Treated and untreated balsa blocks after exposure to *Poria monticola* in jar tests.

Tow row, left to right:

No. 3, untreated.—Block is badly shrunken and crumbling.

No. 10, untreated.—Block is slightly overgrown with mold, which has apparently prevented growth of the test fungus, though some mycelium of the latter was apparent on the bottom.

No. 14, untreated.—Extensive decay is apparent throughout, especially on the lower portion. The extent of the softening which occurred is indicated by the thumb dent visible in the photograph.

Lower row:

No. 15, untreated.—Similar to No. 10 in that mold has completely suppressed growth of the test fungus.

Nos. 1 and 9, treated with chlorinated phenol preservative solution.—Both blocks remained bright and free from either decay or mold.

FIGURE 9.

Treated and untreated balsa blocks after exposure to *Lenzites trabea* in test jars.

Top row, left to right:

No. 4, untreated.—Specimen crumbling under slight pressure.

No. 11, untreated.—Decay has progressed nearly as far in this block as in No. 4. Abortive period fruiting bodies of the fungus visible at top.

No. 20, untreated.—Bottom of block greatly softened and shrunken.

Lower row, left to right:

No. 22, untreated.—Extensively decayed throughout as evidenced by thumb dent at bottom.

Nos. 5 and 13, treated with chlorinated phenol preservative solution.—Both blocks are still bright and free from decay.

to determine the effectiveness of a 3-minute dip in chlorinated phenol preservative solution in preventing decay by the same fungi, using the soil-contact method of Leutritz (10). Wide-mouthed, screw-top jars 8 cm. in diameter by 19 cm. high and of 970 ml. capacity were used. In each jar was placed 200 gms. of a clay loam woods' soil mixed with 100 ml. of water, this amount sufficing to make the soil cohere in a soft ball when squeezed in the hand. After firming and leveling the soil, a balsa feeder block 1½ in. square by 7/16 in. thick, dipped in hot Bacto-wort agar to make a better nutrient medium was placed in contact with the surface of the soil in each jar. The jar assemblies were then sterilized by autoclaving at 15 lbs. pressure for 20 minutes. Upon cooling, 6 jars were inoculated with *Poria monticola* and 6 with *Lenzites trabea*, and all were stored in an incubator at 78° F.

After the fungi had made a good growth over the surface of the feeder blocks a test block of balsa, measuring 1½ in. square by 4 in. long, was introduced into each of 4 jars with the fungus. The blocks used were selected to show a considerable range in density. Prior to introduction into the jars the blocks were oven-dried and weighed, after which they were submerged in a pan of water and autoclaved at 5 pounds pressure to kill extraneous molds that are commonly present and to provide a good moisture content. One of the blocks was stood on end on the feeder block in each jar. Four other blocks were oven-dried, weighed, cooled, given a 3-minute dip in the preservative solution and the weights recorded after draining. After standing for 3 days to allow the volatile vehicle to evaporate, these blocks were submerged and soaked in sterilized water to provide a good moisture content, and were stood on end on the inoculum blocks in the remaining 4 jars. Thus, in the set of 6 jars with each fungus there were 4 with the untreated blocks and 2 with treated ones. The jars were then incubated for 3 months at 78° F. After this exposure the blocks were removed, examined, oven-dried to a constant weight, and reweighed.

Throughout the 3-month incubation period *Poria monticola* grew well on all the feeder blocks, which were completely softened and decayed. It also grew well on two of the four untreated controls but failed to develop or made but scanty growth on the other two, which became somewhat overgrown by a mold. Of the two blocks on which the test fungus grew well, one was badly

shrunken and crumbled to bits when handled; the other, although less shrunken, was extensively decayed and so softened as to be readily indented upon pressure with the fingers. The untreated control blocks showed percentage weight losses on an oven-dry basis amounting to 61.0, 0, 40.1, and 2.4, respectively. Both the treated blocks remained clean and bright and free from evidence of decay or mold. Dry weights of the treated blocks were calculated as in the case of the block assemblies described previously but neither showed weight loss. The large weight losses sustained by the two untreated blocks on which *Poria microspora* grew well testify to the effectiveness of this fungus as a destroyer of balsa. The effectiveness of the preservative treatment in preventing both attack by this fungus and the growth of molds is also demonstrated.

The jar tests with *Lenzites trabea* proved much more satisfactory in that in no instance did a mold interfere with the growth of the test fungus. All the untreated controls, as well as the feeder blocks, were attacked and covered with a luxuriant growth of the test fungus at the end of the exposure period. As in the case of the *Poria monticola* series, however, both the treated blocks were clean and bright and neither showed any evidence of attack during this time. The extent of the attack on the controls appeared to vary in approximately inverse proportion to the density. The lightest of these was almost completely destroyed and exhibited a heavy poroid layer of the fungus fruiting at the top. The next heavier, which also had a similar layer fruiting at the top, crumbled readily, while the remaining blocks were softened and shrunken at the base but showed less evidence of decay at the top and only a slight poroid layer at this point. These showed percentage weight losses on an oven-dry basis amounting to 68.8, 41.9, 23.4, and 30.8, respectively. The calculated weight losses in the treated blocks were so small as to be insignificant when the method of calculation is considered. The effectiveness of the treatment is even more clearly demonstrated in this instance than with the preceding test fungus.

Effect of the Preservative Treatment on the Shear Strength of Glue Joints.

In order to determine the effect of the preservative treatment on glue joints in balsa a number of shear tests were made on

joints prepared in treated and untreated blocks. For these tests 144 blocks 2 × 1¾ × ¾ in. were selected within the density range and divided into sets of 18 in each of 8 density ranges as follows: 8 to 8.4; 9.0-9.4; and so on up to 11.5 to 12 lbs. per cu. ft. This was done to assure uniform distribution over the entire range. The two lightest blocks were then paired and placed in group 1, the next two heavier were paired for group 2, the next two for group 3; the next pair were again placed in group 1 and the process repeated until the entire lot had been subdivided into three pairs in each of three groups in the 8 density ranges. One 2 by 1¾ in. side of each block was sanded to provide a smooth gluing surface.

First group (*untreated controls*).—The sanded face of each block of each pair was coated with urea-formaldehyde glue, using a 2-in. paint brush, and the glued faces were placed in contact. After all blocks of group 1 had been glued they were placed in a press and a pressure of 25 lbs. per sq. in. was applied, as required by the specification for life floats (8). Since the room temperature was approximately 80° F., this pressure was maintained for 6 hours as directed by the schedule given in the specification. After removal from the press the glued blocks were stored one week and were then tested for shear strength by the method given in the specification for urea-formaldehyde glue (13).

Second group (*treated—24-hour drying period*).—All blocks of this group were given a 5-minute dip in Type B preservative solution, after which they were dried under room conditions 24 hours. They were then glued and tested, in the same order as the first group, except that, since the room temperature was 90° F., the time in the press was reduced to 4 hours.

Third group (*treated—48-hour drying period*).—The blocks were treated exactly as were those of the second group except that the drying period after preservative treatment was 48 hours.

A summary of the results of the shear tests on glue joints in untreated balsa and on joints in balsa given a 5-minute dip in chlorinated phenol preservative, with 24-hour and 48-hour drying, is given in Table 1.

The wide variation in the results for both treated and untreated specimens is apparent. In the case of the untreated blocks this

TABLE I.*
RESULTS OF SHEAR TESTS ON GLUE JOINTS IN TREATED AND UNTREATED BALSA BLOCKS.

Density range of blocks (lbs. per cu. ft.)	Untreated blocks (lbs.)	Ave.	Treated blocks 24-hr. drying after dip (lbs.)	Ave.	Treated blocks 48-hr. drying after dip (lbs.)	Ave.
8.0- 8.4	430 600 710	580	385 200 270	285	640 330 410	460
8.5- 8.9	610 570 340	507	685 470 570	573	550 400 510	487
9.0- 9.4	400 (decayed wood) 750	575	460 750 710	640	200 670 400	423
9.5- 9.9	530 950 810	763	710 670 450	610	400 350 230	327
10.0-10.4	750 430 490	557	370 460 340	390	200 240 240	227
10.5-10.9	590 620 740	650	520 780 450	583	470 820 970	753
11.0-11.4	540 665 735	647	440 580 480	500	530 230 340	367
11.5-12.0	600 910 585	698	640 390 750	593	200 580 710	497
Overall average		624		522		443

* The author wishes to express his appreciation to Messrs. T. DiMasi and L. Ritter for conducting the glue tests reported as part of this work, and to Mr. D. F. Richardson for helpful suggestions in connection with the presentation of the results in the official report previously issued (15). The opinions expressed herein are those of the author and do not necessarily reflect the official views of the Navy Department.

variability is difficult to explain since most joint failures occurred in the wood rather than in the glue, and the variation bears no apparent relationship to the density of the wood. In the case of the treated blocks it is unfortunate that no records were kept of the amount of preservative absorbed; it is quite possible that some correlation might be demonstrated between the preservative retention and the joint strength. A further fact for which no explanation can be offered is that the blocks permitted to dry for 48 hours before gluing showed a lower overall average joint strength than those dried for only 24 hours. In view of these

considerations the results can in no way be considered conclusive; however, they indicate a tendency for treatment with the preservative solution used to reduce the strength of glue joints. This is in agreement with data published (9; 2) in which personnel of the Forest Products Laboratory reported the results of experiments on the gluing of wood treated with various preservative solutions. In these experiments it was found that treatment with solutions containing water-repellent ingredients such as are required by the specification (3) for Type B preservative, markedly interferes with the adhesion of glues. This interference is directly attributable to the water-repellent ingredients; treatment with solutions of chlorinated phenols alone had no apparent effect on adhesion. From these results it appears that balsa may be treated with chlorinated phenol solution of the kind used except for the omission of the water-repellent ingredients, and that gluing of the treated wood then will present no more problems than are normally present in life float manufacture.

Effect of the Preservative Treatment on the Weight and Buoyancy of Balsa Life Floats.

Treatment of wood with chlorinated phenols will obviously increase its weight slightly and will correspondingly decrease the buoyancy of life floats made from it. The extent to which this will occur will depend upon the absorptiveness of the wood, which, as has been discussed, is extremely variable for balsa. Of the series of 76 blocks used in the dip tests, 15 which ranged in density from 8 to 12 lbs. per cu. ft. showed an average weight of 24.2 gms. and an average absorption of preservative solution of 7.2 gms., or an amount equaling 30% of the average weight. Assuming anon-vola tile content of 15% for the solution, this is equivalent to a 6 % increase in weight of the dried specimens. However, omission of the water-repellent ingredients would reduce the non-volatile content of the solution to 5%, in which case the weight increase for a similar absorption would have been only 1.5%. On this basis, a 3-min. dip treatment of the wood for a 25-person life float weighing about 200 lbs. would only increase its weight by about 3 lbs. However, since the absorption of the preservative solution by large pieces of wood would be much less proportionately than for small blocks, the actual

weight increase of a life float would be less than 3 lbs., which would not be considered excessive. Reduction in buoyancy would be by the same absolute amount; this would be much less proprotionately since the average buoyancy for this size float is approximately 1000 lbs.

Recommendations.

A simple dip treatment in the preservative solution is considered adequate for new life float construction and should not add greatly to the cost. While a 3-minute dip was used for most of the small specimens tested in these investigations, for the larger pieces used in life float construction a 5-minute dip is recommended to allow a greater margin of safety. In the construction of new life floats it is therefore recommended that the applicable specification be revised to require that all pieces of balsa entering into floats be given a dip treatment, after all possible cutting and milling operations have been performed but prior to any gluing operations, with a 5% solution of chlorinated phenol. The treating solution shall conform to the requirements for Type B of the Navy Department Specification 52W5, except that it shall contain no water-repellent ingredients. Treatment shall consist of completely immersing the pieces for 5 minutes at room temperature, after which they shall be removed and allowed to dry at least over night before gluing. After final shaping and sanding, the finished float body should be given a similar dip to make sure that all exposed surfaces are treated. For this purpose the water-repellent ingredients are not objectionable but highly desirable, and the preservative should be used in complete conformity with the specification requirements. The durability of the canvas covering could also be greatly increased by treatment with an effective preservative.

It is further recommended, in view of the rapid deterioration to which balsa life floats are subject, the uncontrollable variability of the wood in specific gravity and other physical properties, and the difficulty of obtaining it in event of war, that a study be made of the possibility of using some synthetic material for float construction, such as a foamed plastic of suitable properties, which would be nonabsorptive and not subject to decay.

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NORFOLK NAVY YARD

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PART II.

(Reprinted Articles)

The articles contained in this Part II represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the Journal, for articles originally published elsewhere which contribute to the purpose of the Society: to further the advancement of naval engineering.

GERMAN U-BOAT DESIGN AND PRODUCTION.

This article by Comdr. J. F. Starks, R. C. N. C., which is an abstract of a paper read before the Institution of Naval Architects, is reprinted from the Journal of the Royal United Service Institution for May, 1948.

The story of the development of Germany's U-boat force is a clear example of a major maritime Power devoting at first the main part, and later practically its entire naval effort to one class of warship, designed to operate chiefly against merchant shipping.

Comparison between British and Allied submarine design and production and that of the German U-boats is inevitable. It is difficult, if not impossible, to present a fair tabular comparison between designs of various countries, but it is thought not inappropriate to refer to this question.

In the first place it should be stated that there has been, and still is a tendency in many quarters to associate the submarine with the German nation, not an unnatural association in view of the severe effects of German submarine warfare on British ships and communications. This had often led to the conclusion that, if a submarine or piece of submarine equipment be German, then it is of necessity of the highest quality in all respects. This view, it is suggested, has been over emphasized.

It is important to appreciate that the design of a submarine and any innovations introduced are influenced, as is any engineering project, by the work it has to do and, in the case of a warship, the opposition it may have to encounter. If all goes well with a design it is not changed merely for the sake of change, particularly in war.

Regarding comparison with British submarines in particular, it is worthy of note that Great Britain has been in the past most definitely an anti-submarine nation, while Germany has been a pro-submarine nation. This fact alone is enough to show the virtual impossibility of anything but a broad comparison between German submarines and those of British design. If the Germans outshone the British and Allied designs it was rather in the

implementing of new developments than in their conception. In fact, on reviewing the German submarine trend, it is surprising to see how parallel were the lines of thought of the Allies and the Germans, but in view of the enormous effort the latter allocated to submarines they were able to introduce the alterations into their U-boats much more quickly. Moreover, the opposition, targets and theatres of operation for the Allies were so very different from those of the Germans. The latter's main effort was in the Atlantic, with comparatively close bases. The theaters for the Allies' submarines were chiefly the Mediterranean and the Far East at very great distance from their bases.

Apart from the developments dictated by war lessons, the amount of effort devoted to the submarine design in different countries is, of course, most important. If further evidence is required of the magnitude of the German submarine effort from that already demonstrated by the Battle of the Atlantic, it is hoped that the data given in this paper will provide it.

Although the numbers of submarines employed and the strength of Allied forces must of necessity affect the outcome, it is a tribute to the early war-time designs of U-boats that they withstood for so long the full onslaught of the Allied anti-submarine forces. Their toughness was shown on numerous occasions, and the U-boats, for the second time in history, nearly won.

It is suggested that a fair summarized conclusion of German U-boat design is:—

(*a*) The submarines actually employed on operations had by no means outmoded those of the Allies. Each country had submarines capable of fulfilling the duties required of them.

(*b*) The modifications made to the war-time U-boats, in particular the *schnorkel*, were in the main *defensive* in character. They were not outstanding inventions; they employed well-known submarine principles and the new devices were fitted because the theaters of operations and efforts of the Allied anti-submarine units forced the Germans to do so.

(*c*) At the cessation of hostilities the Germans had submarines almost available and designs projected that were to set a new fashion in submarine warfare. These new types of submarine were the best illustration of the tremendous efforts given to U-boat development and had they arrived on operations they

would have been no small "headache" for the Allies. Had they come earlier they might have prolonged the War.

(*d*) Lastly the Type VII C U-boat, which comprised some 50 per cent of the German submarine fleet, was "a very good submarine of a conventional type."

Stages in Development.

Stage I. The Submersible or Conventional Submarine.

The Germans did not officially begin designing their own U-boats until the early 1930's, but prior to this they had by no means ignored the question of submarine design. They had their interests and experts in several submarine design and building projects in a number of countries outside Germany and in 1931, approximately, were able to commence the build-up of a U-boat service with a nucleus of well-experienced design staff. There is no doubt, too, that their factories and shipyards had taken a keen interest in production of submarines and auxiliaries.

The trend of design from Type I to Type IX was steady and consistent. The early designs were not very different in principle from those that operated towards the close of the 1914-18 War, i.e., with moderate surface speed and endurance and the usual 1-hour endurance at a full submerged speed of approximately 9 knots. The submarine was intended to patrol, dived by day and surfaced by night.

As their designs and national ambitions developed, so the surface endurance increased. Similarly the conception of the method of closing and attacking convoys *on the surface* by night led to the demand and supply of increased surface speed. This trend culminated in the Type IX D2 where a surface endurance of 30,000 miles at 10 knots and a top surface speed of 19.2 knots was achieved.

It is of interest, and most important in comparing U-boats with submarines of other navies, to note that the high surface endurance and surface speed were achieved primarily by:—

(i) Stowing oil fuel in main tanks, thus reducing the reserve of buoyancy and sea-keeping qualities of the ships on the surface.

(ii) Adopting the practice of allowing the engines to be overloaded for short periods to get the maximum surface speed. This feature, which would of course be adopted by any ship in an

emergency, came to be an accepted practice with the German U-boat.

(iii) Keeping accommodation requirements to the bare minimum.

The two most typical examples of the early efforts up to 1943 were the Types VII C and IX D2. These designs were, of course, subject to variations, in particular the gun armament was increased as war experience was gained. This latter is best shown by reference to the stages in bridge design for the Type VII C. As an example of this stage, remarks on the Type VII C's main features are given at the end of this section.

Before proceeding to the next stage in development, mention must be made of the various incidental problems which inevitably arise in wartime, such as minelaying submarines, supply submarines, cargo-carrying submarines, etc. The types produced are given elsewhere in this paper and it is sufficient to say at this stage that, in general, the Germans adopted the procedure, as have most countries, of adapting an existing design for such special purposes. Requirements such as minelaying and supply submarines are usually, though not always, a passing phase dictated by special operational needs, and the policy of using a parent design for such ships is considered sound.

It would have been wise to investigate a completely new design only if large numbers had been required.

Stage II. The Schnorkel.

The next stage in development was the fitting of the *schnorkel.*

Much has been published of this so-called German invention. The circumstances bringing about its introduction were that the Germans had achieved finality in the design of the submersible. The latter was now fast, tough, well armed, and had wide range, *but* it had to surface at regular intervals to charge its batteries and air reservoirs.

The U-boats had extended their scenes of operations over most of the World and had achieved appreciable success. The main effort was, of course, confined to the Battle of the Atlantic, and by 1942-43 the Allied anti-submarine forces, in particular radar-fitted aircraft, were beginning to take a heavy toll of the U-boats. Too many, from the German viewpoint, were being caught on the surface, even by night. This led to the requirement that the

submarine should not surface and hence some means had to be provided to enable the Diesel machinery to be run while dived, viz., the *schnorkel*.

As is well known, this comprised a mast leading to the surface through which air could be drawn both for the engines and the air compressors and also through which the engines could exhaust. More details of it are given later.

It can be fairly said that this device was not a purely German invention. Prior to the War most countries possessing submarines had investigated such a scheme, in particular the Dutch. Both Britain and Germany began serious development of this project in 1940 but both discarded it *pro tem* as the need for its introduction had not become vital. Need for further development during the War did not arise for the Allies, but for the Germans it most certainly did. Thus it was that, in 1943, action was taken to fit all existing U-boats with the *schnorkel*. All later boats were also to be fitted.

There are many problems associated with *schnorkelling* submarines, not the least being physiological, but the Germans achieved a good measure of success, some of their U-boats staying at sea for several weeks without surfacing.

Stage III. Need for High Submerged Speed.

While the *schnorkel* had, to some extent, countered the Allied anti-submarine effort, the U-boats were by no means free of tribulations, and the Germans wisely regarded the *schnorkel* as purely a stop-gap. By fitting it they had changed their U-boat fleet from a very mobile force into a much more static one, the top speed using the *schnorkel* being of the order of 6 knots. The U-boats could, of course, operate more closely to enemy coasts with less fear of detection than hitherto, but their movement in the wide areas of the Atlantic was severely restricted, as was their ability to make contact with the enemy. Moreover, at this stage of the War, having carried out an attack it was not possible to surface at night and escape at high speed without being detected and attacked by aircraft and/or surface escorts.

Later, in 1943, therefore, came the conception of a *schnorkel*-fitted, high submerged speed submarine, submerged propulsion being by batteries and motors. This stage is illustrated by the Types XXI and XXIII, the former being the ocean-going type and the latter for coastal work.

The submarines were designed for continuous submerged patrol. Surface-keeping qualities and main gun armament were surrendered and the hulls streamlined to the greatest possible extent to reduce submerged resistance.

The Type XXI is described at the end of this section as a typical example of this stage.

At this juncture also, the effect of Allied bombing was being felt and this, in conjunction with the need to get this new type into production as soon as possible in large numbers, led to the adoption of the prefabricated scheme, described later.

Stage IV. Approach to the True Submarine.

The Germans fully realized that although large battery capacity combined with streamlining provided a higher submerged speed, the weakness of this type lay in the fact that the top speed could only be maintained for a short time. Moreover, the situation demanded even higher submerged speed. In parallel with the Type XXI, therefore, work was carried out to produce a propulsion unit which would operate with the submarine completely divorced from the atmosphere. Such a unit required large quantities of stored oxygen or an efficient oxygen carrier.

A large number of investigations were conducted, but the only unit which came into production was a turbine plant which used hydrogen peroxide as the method of storing oxygen. This unit gave higher speeds and greater time endurance at high speed than the battery-driven submarine but at the expense of heavy fuel consumption, reduced radius of action at the cruising speeds, and less flexibility.

No such submarine operated in the War, but it was intended that any operating would still be fitted with the *schnorkel*.

The Types XVII and XVIII were classes of submarine of this stage and were the nearest approach to the ultimate aim of the submarine designer, viz., the true submarine, i.e., one which will operate for a whole patrol completely divorced from the air.

The following is a summary of the four stages of development:—

DESIGN PARTICULARS OF THE FOUR STAGES

	Stage I		*Stage II*	*Stage III*		*Stage IV*
Summary of design features	High surface speed, long range of operation		Fitting of *schnorkel* to existing types. The U-boats had lost the effect of their high surfaced speed and endurance	Surface qualities subordinated to those submerged. *Schnorkel* included in original design		Increased submerged full speed and endurance. *Schnorkel* included in design
	VII C	*IX D2*		*XXI*	*XXIII*	*XVII B*
Displacement—						
Surfaced	769	1,616	Maximum submerged speed while using the *schnorkel* was of the order of 6 knots	1,621	232	312
Submerged	871	1,804		1,819	256	337
Speed—						
Surfaced	17.3	19.2		15.6	9.7	8.5
Submerged	7.6	6.9		15-16	10	25*
Endurance—						
Surfaced	9,700 miles at 10 knots	32,300 miles at 10 knots		15,500 miles at 10 knots	1,350 miles at 9.7 knots	3,000 miles at 8 knots
Submerged	80 miles at 4 knots	57 miles at 4 knots		365 miles at 5 knots	175 miles at 4 knots	40 miles at 4.5 knots (on batteries)

* German estimate of speed with two turbines fitted.

Type VII C and VII C (41).

A diagram of the layout is given in Fig. 1.

The Type VII C was designed along conventional lines, being the usual saddle tank submarine, armed with torpedoes (and/or mines) and gun, fitted with two periscopes, direct Diesel drive on the surface, main motors for submerged propulsion with two battery sections. As has been stated, the *schnorkel* was fitted to this class during the course of the War. Some of its main features were:—

Hull.—The pressure hull, except for the closing plate over main machinery, was all welded and circular sectioned throughout. The plating was of a carbon content steel of approximately 0.21 per cent and ⅞ in. thickness amidships, decreasing somewhat towards the end of the vessel.

The frames, welded internally approximately 23 in. apart, were of a bulbous "T"-shape section toe welded to the hull.

The dome bulkheads at the ends of the hull were steel castings.

External tanks were also all welded.

Main Tanks.—The position of the tanks was, as shown in Fig. 1, the most striking feature being the internal No. 3 main tank. It is considered that this internal main tank, with its very large hand-operated kingston flaps, was an undesirable arrangement and detracted somewhat from the integrity of an otherwise very tough hull.

Main vents were hand-operated from the control room. There appears a very real risk of the shafting led fore and aft to the end main tanks being distorted under depth-charge attack.

Main tanks were blown by using H.P. air to bring the submarine awash and the blowing completed by using the exhaust from the main Diesel machinery.

Speed.—The VII C U-boat had a full speed of 17.7 knots in the surface condition without fuel in the main tanks, the continuous full speed being 17.3 knots.

Submerged full speed was 7.6 knots, and this could be maintained for one hour.

Armament. (i) *Torpedo.*—There were four internal torpedo-tubes forward and six reload torpedoes. Aft there was a single internal torpedo-tube, with one reload.

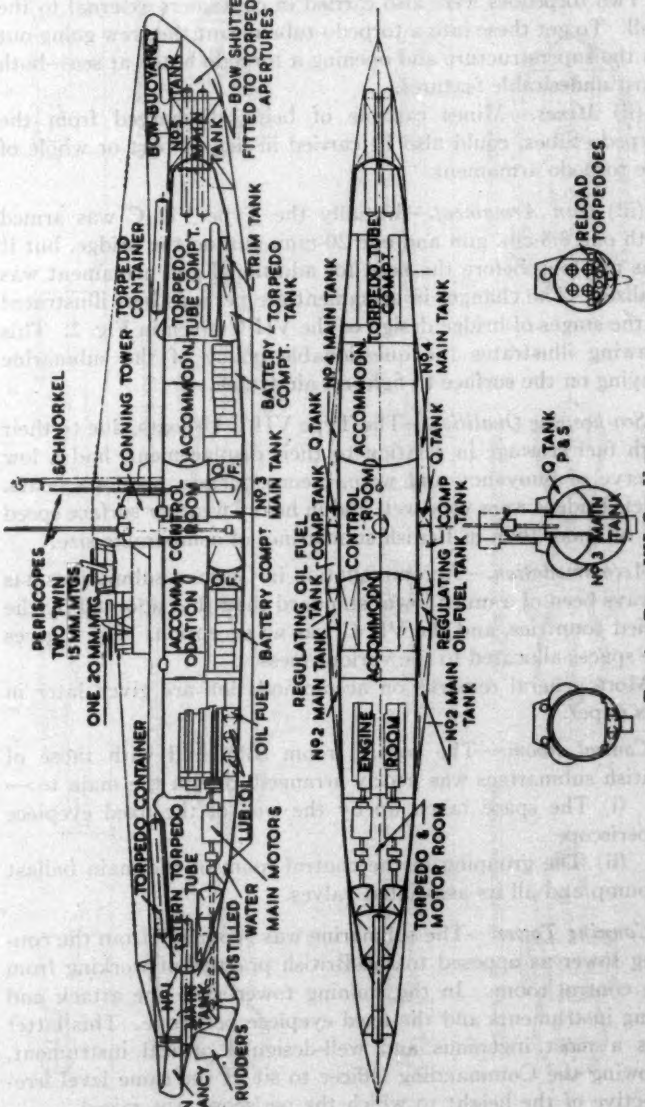


FIG. 1.—TYPE VII C. U-BOAT

Two torpedoes were also carried in containers external to the hull. To get these into a torpedo-tube meant the crew going out on the superstructure and opening a torpedo hatch at sea—both most undesirable features.

(ii) *Mines.*—Mines capable of being discharged from the torpedo-tubes, could also be carried in lieu of part or whole of the torpedo armament.

(iii) *Gun Armament.*—Initially the Type VII C was armed with one 8.8-cm. gun and one 20-mm. gun on the bridge, but it was not long before the need for additional gun armament was realized. The changes in armament are perhaps best illustrated in the stages of bridge design of the VII C given in Fig. 2. This drawing illustrates the questionable policy of the submarine staying on the surface to fight off air attack.

Sea-keeping Qualities.—The Type VII C U-boats, due to their high fuel stowage in relation to their displacement, had a low reserve of buoyancy and were, inconsequence, poor sea boats. Their bridges were very wet, and in heavy weather surface speed fell off more than in British submarines of comparable size.

Accommodation.—Accommodation in German submarines has always been of a much lower standard than that adopted by the Allied countries, and the VII C was no exception. Fig. 1 gives the spaces allocated to the various messes.

More general remarks on accommodation are given later in this paper.

Control Room.—The control room compared with those of British submarines was poorly arranged, due in the main to:—

(i) The space taken up by the well of the fixed eyepiece periscope.

(ii) The grouping in the control room of the main ballast pump and all its associated valves.

Conning Tower.—The submarine was "conned" from the conning tower as opposed to the British practice of working from the control room. In the conning tower were the attack and firing instruments and the fixed eyepiece periscope. This latter was a most ingenious and well-designed optical instrument, allowing the Commanding Officer to sit at the same level irrespective of the height to which the periscope was raised.

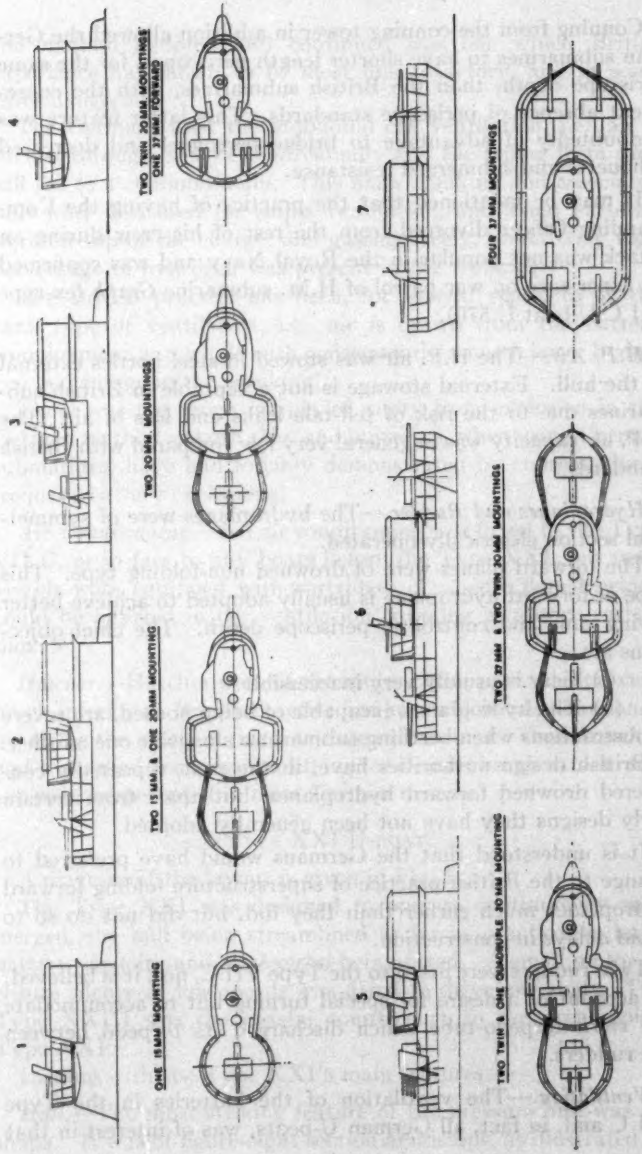


FIG. 2.—DEVELOPMENT IN GUN ARMAMENT FOR TYPE VII C U-BOAT AND STAGES IN BRIDGE DESIGN

Conning from the conning tower in addition allowed the German submarines to have shorter length periscopes, for the same periscope depth, than the British submarines, with the consequent absence of periscope standards. This latter feature was undoubtedly of advantage to bridge personnel and decreased silhouette and submerged resistance.

It may be mentioned that the practice of having the Commanding Officer divorced from the rest of his crew during an attack was not popular in the Royal Navy and was confirmed by experience on war patrol of H.M. submarine *Graph* (ex-type VII C U-boat U.570).

H.P. Air.—The H.P. air was stowed in steel bottles external to the hull. External stowage is not acceptable in British submarines due to the risk of tell-tale leaks and loss of air. The H.P. air capacity was in general very low compared with British standards.

Hydroplanes and Rudder.—The hydroplanes were of symmetrical section electrically operated.

The forward planes were of drowned non-folding type. This type of forward hydroplane is usually adopted to achieve better diving time and control at periscope depth. The chief objections are:—

(i) Gear is usually very inaccessible.

(ii) The hydroplanes, incapable of being housed, are severe obstructions when berthing submarines alongside one another.

British design authorities have, in the past, repeatedly considered drowned forward hydroplanes, but apart from certain early designs they have not been generally adopted.

It is understood that the Germans would have preferred to change to the British practice of superstructure folding forward hydroplanes much earlier than they did, but did not do so to avoid delays in construction.

Twin rudders were fitted to the Type VII C, not, it is believed, on account of a desire for special turning but to accommodate the stern torpedo-tube which discharged its torpedo between the rudders.

Ventilation.—The ventilation of the batteries in the Type VII C and, in fact, all German U-boats, was of interest in that

the method adopted and continued was one which British experience has shown to be most unsatisfactory, and in some cases dangerous.

The Germans used the individual cell ventilation, i.e., air is sucked through each cell individually and the piping from each cell led to a common main. This main trunking and associated fans were also used for ship's ventilation, and when batteries were on top of the charge, and gassing freely, a relatively high percentage of hydrogen was present in the trunking.

The British practice has been, for several years, to fit the tank type of ventilation, i.e., air is drawn from the battery compartment as a whole with consequently a much lower hydrogen percentage in the trunking.

There was not much margin of safety from explosion in the German method as both they and ourselves, when using German submarines, have had forcibly demonstrated by comparatively frequent battery explosions.

Air Conditioning.—No air conditioning was fitted to the Type VII C, or in fact to any boats before the Type XXI, but most vessels were equipped with cartridges of caustic potash which could be inserted in the ventilation trunking to absorb carbon dioxide.

Hatches.—Hatches were of the spider clip type—a design very much inferior to British practice and one which was abandoned by us for the whole of our war-time submarines. These hatches had a marked tendency to lift under depth-charge attack and yet were perpetuated in most German designs.

Type XXI U-boat.

A diagram of the layout is given in Fig. 3.

The Type XXI was designed to operate continuously submerged, the hull being streamlined as far as practicable, large battery capacity and a *schnorkel* being fitted. A small motor to ensure silence at low speeds was fitted to drive one shaft. The Type XXIII was the coastal counterpart to the ocean-going Type XXI.

Dealing with the Type XXI's main features:—

Hull.—The most striking feature of the pressure hull was its shape. It was of figure-eight section amidships, as illustrated in

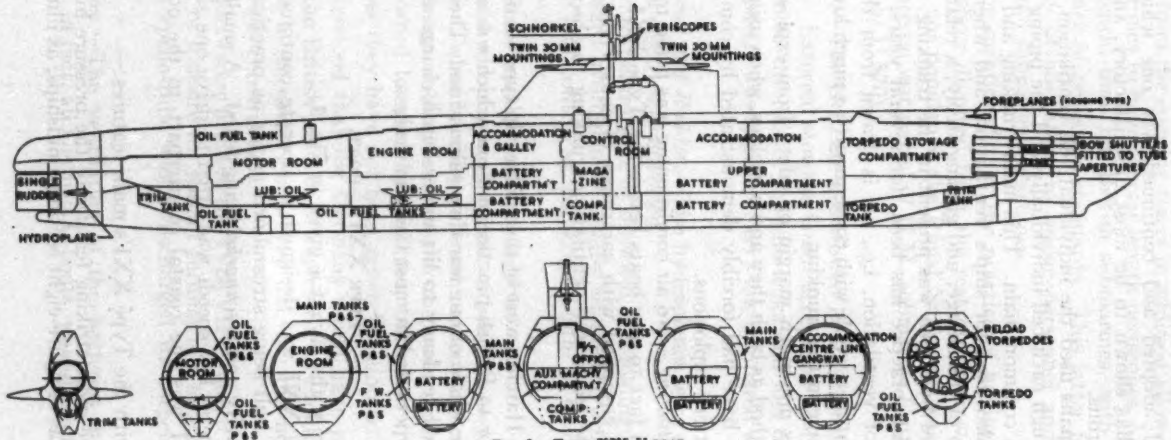


FIG. 3.—TYPE XXI U-BOAT

the diagram. The departure from the conventional circular section, which had been consistently maintained in all previous designs, was to obtain the most economical stowage of the large battery capacity.

It will be seen that the battery sections were arranged two-tier fashion, one compartment over another.

As described elsewhere, the hull was built in eight sections and when welded together there was no means of getting certain large items of equipment out of the boat without cutting the submarine in two, or cutting out a prohibitive amount of plating and moving other machinery.

The hull was of similar steel to that used for the Type VII C and was welded throughout, except where steel castings were fitted, which were riveted.

The frames were welded *externally* over the top part of the section throughout the greater part of the length. This was to simplify prefabrication and to obtain clear runs of cable, piping, etc. At the same time it certainly facilitated accommodation of equipment inside the pressure hull.

The forward dome bulkhead and part of the torpedo-tubes consisted of a complete casting. The after bulkhead was flat as opposed to the usual dome shape.

Speed.—The surface speed was of the order of 15½ knots. The whole object, however, was to obtain maximum submerged speed, and all other requirements were subordinated to this requirement.

The submerged speed was claimed by the Germans to be 16 knots, but there is reason to believe that the top speed for one hour's endurance would have been much nearer 15 knots.

Diving Depth.—The hull plating over the largest section was approximately 1⅛ in. thick and a practical diving depth of 500 ft. is considered a reasonable figure.

Armament.—There were six bow torpedo-tubes with the impressive reload capacity of a further 17 torpedoes. No stern tubes were fitted.

The torpedo stowage compartment, because of the large middle section of the submarine, was of "cathedral-like" dimensions and

contained elaborate arrangements for loading and stowing torpedoes.

Gun Armament.—No large gun was fitted as it would give rise to undue submerged resistance.

A concession to the gun-minded authorities led to the fitting of two machine-gun mountings on the bridge. These were well protected against small arms fire from aircraft or surface ships.

Schnorkel.—A periscope *schnorkel* mast was fitted. This mast was very slow in raising and lowering and in this and other respects was not entirely satisfactory.

Sea-keeping Qualities.—As this class was designed for operation submerged, sea-keeping qualities were sacrificed to submerged qualities. As a result the bridge was almost untenable in heavy weather, and British experience with these ships was that it was necessary to batten down and run on the surface with the *schnorkel* mast up.

Bow and stern buoyancy tanks were fitted, however, but did tittle to mitigate these conditions.

Accommodation.—Accommodation was again of much lower standard than in British submarines. The center-line gangway necessitated the messes being disposed on each side of the ship.

There were only three W.C.s and three washbasins for the entire crew.

H.P. Air.—The Type XXI saw the fitting of H.P. air operation to main vents, torpedo-loading winch, *schnorkel* mast-raising gear, and the capstan.

The H.P. air capacity was even further reduced over that in previous designs on the basis that the boats would rarely surface.

Hydroplanes and Rudder.—The hydroplanes and steering gear were telemotor operated—the first time the Germans had adopted this method, presumably with a view to keeping noise to a minimum.

A feature which cannot be recommended was that the entire telemotor units were external to the hull. There is real danger of oil track and of ingress of sea water into the telemotor system.

The forward hydroplanes were rigged from the superstructure, as opposed to drowned hydroplanes in earlier types.

A single rudder was fitted.

Telemotor System.—The Type XXI had the most ambitious telemotor system of any German U-boat. It was designed to operate the hydroplanes, steering gear, periscopes and bow caps, using two pumps worked in conjunction with a ramless air-loaded accumulator.

This system gave repeated trouble, probably due to inferior workmanship, and teething troubles generally, and was one of the main causes of no Type XXI being ready for patrol when the War ended.

In contrast British submarines rely to a much greater extent on the telemotor system and it has proved generally very reliable.

Ventilation.—Battery ventilation was again of the individual cell type. Hydrogen disposal units were originally fitted but were not satisfactory and were removed.

Air Conditioning.—For the first time the Germans fitted air conditioning proper to a submarine in the Type XXI. It was possible to cool, dry and heat the air as required. It is certain that with the wide range of theaters of operation in all climates some of the earlier boats had been forced to surface to clear the air and were subjected to attack while doing so.

Type XVII and XVII B U-boats.

The special feature of the Type XVII was its turbine machinery, permitting submerged operation without contact with the atmosphere.

The main features of the Type XVII B were:—

Hull.—The hull was of circular section with underslung tanks, giving a diving depth of 330 feet. The displacement was 312 tons surfaced.

Speed.—Surface speed was estimated at approximately 8 knots.

The submerged speed using the turbine was never, as far as can be ascertained, accurately measured. The Germans claimed a top submerged speed of 25 knots, with two turbine units. It is believed that in actual fact only one turbine was finally fitted so the speed must have been appreciably less than 25 knots.

Armament.—Two bow torpedo-tubes with two reloads.

Schnorkel.—A periscopic *schnorkel* was fitted.

Accommodation.—The standard of accommodation was very low indeed and the submarine could have been used only for a very brief patrol.

There was little else to comment on in the Type XVII B and although fitted with torpedo-tubes it is most probable that this class was designed chiefly as a forerunner of larger turbine fitted submarines. The first four models (Type XVII) were, in fact, purely experimental submarines.

PRODUCTION.

A type number was given to each design that was given serious consideration. There were in addition many other designs which did not progress to the stage of being given a type number.

It is a good illustration of the effort devoted to submarines that the Germans, from the early 1930's onwards, investigated and produced designs for some thirty classes of submarine with as many as four to six different variations of approximately ten of these types. These figures do not include certain classes of midget submarines produced towards the end of the War, nor can these statistics give any idea of the range and variety of separate investigations progressed. Not all the types were actually built, but of the thirty or more, some twelve parent types, with variations of some, reached the building stage.

As a further example of their exploration of submarine design, details have been obtained of what can at present be described only as a dream—a submarine which could fly! A preliminary investigation of this project, which, as far as is known, did not progress beyond the drawing board, although it is reported to have excited keen interest in high places, led to the not unexpected conclusion that it would not fly. With drastic modifications it could probably have been made to fly but would then have become almost a normal flying boat and would have been unable to submerge (voluntarily).

In all, some 1100 U-boats were completed during the War, excluding midget submarines and the numerous hulls that were ruthlessly scrapped as another type was developed. The numbers of the various types built were:—

Type	Numbers built	
	Before the War	*During the War*
I A	2	0
II A	6	0
II B	18	2
II C	6	2
II D	0	16
VII	10	0
VII B	8	16
VII C	0	572
VII C (41)	0	87
VII D	0	6
VII F	0	4
IX	7	1
IX B	0	14
IX C	0	54
IX C (40)	0	87
IX D1	0	2
IX D2	0	29
X B	0	8
XIV	0	10
XVII	0	4
XVII B	0	3
XXI	0	120
XXIII	0	61
	57	1098

Total = 1155

These figures are most striking when one realizes the amount of equipment and detailed design in a submarine, and are impressive even though it was the country's main naval effort.

Prior to 1943 production of U-boats was on more or less conventional lines as in Allied countries. Some sixteen firms were engaged on construction and they were controlled by the German Admiralty, the latter preparing the basic designs.

The time to produce a Type VII C or Type IX U-boat was some 7 to 9 months from laying down—not an over-impressive performance when compared with some of the efforts in this Country and taking all factors into consideration.

Prefabrication.

In 1943, as previously stated, the Germans had two problems:—

(i) To counter the A/S forces by adopting a submarine design which could operate submerged for the whole of its patrol and possess reasonably high speed, viz. Types XXI and XXIII.

(ii) To introduce these new submarines into operational theaters at the earliest possible moment—no small problem with Allied air raids getting more and more frequent.

The idea of prefabrication on a national basis was conceived, and was received with much opposition in some quarters, particularly by the Main Shipbuilding Committee. Despite this opposition it was decided to adopt the scheme for the Type XXI and XXIII and any subsequent types. Because of the mixed reception, the whole program was placed under the control of the Speer Ministry of War Production.

The advantages claimed for the prefabrication scheme were:—

(*a*) Dispersal of the sections; desirable in view of Allied bombing.

(*b*) A larger labor force could be employed.

(*c*) Increased production by employment of firms on repetition work.

(*d*) Firms not necessarily shipbuilders could be employed.

A Central Design Agency was set up under the leadership of a German Naval Constructor. This Agency produced the detailed designs of the Type XXI and XXIII, specially allowing for sectional construction and produced detailed drawings, specifications, material orders, etc., based on preliminary guidance drawings provided by the Admiralty.

Despite the overall control of the Speer Ministry the latter's only real function was to ensure the necessary priority of supply of labor and materials.

As applied to Type XXI, the idea was to complete the eight sections entirely with machinery, fittings, pipe and cable leads, etc., all rigged before the sections were brought together; the hull then welded, pipes connected, etc., and the submarine brought to a state ready for launch.

Building Yards.

Three types of building yards were engaged:—

The first did the hull work proper, i.e., built the sections of pressure and external hulls, bulkheads, and certain hull fittings. These firms received the plating and frames already cut and set to shape by the steelmakers. Both shipyards and structural engineering firms—some 32 in all—were engaged in this type of work.

The second stage was for the hull sections to be transported to other firms—some 16 in number—where all wiring, piping, main and auxiliary machinery were installed. Batteries were not installed until just before completion of the whole submarine.

The third stage was for three building yards to assemble the completed sections, which were welded together, on the building slips, carry out all final tests and deliver the submarines to the German Admiralty.

The overriding rules were:—

To carry out as much work as possible before the third stage was reached, thus leaving a minimum of work to be done in the final shipyard and keeping the time on the slipway as short as possible. As an example, it was attempted to get bearings for main shafts machined and aligned in the sections before assembly, but this was found impracticable and abandoned.

No modifications were allowed to interfere with the flow of completed sections. This necessitated the closest supervision and even then was not always fully achieved despite an army of inspectors and a multiplicity of instruction sheets and directions.

Transport of Sections.

While some of the hull section yards were not immediately adjacent to waterways, the choice of the various firms was influenced by proximity to canals and rivers. Thus transport was for the most part made by water, special barges and pontoons being made to suit the various sections.

Results of Prefabrication.

Because no Type XXI U-boat carried out a war patrol it may be argued that the prefabrication scheme was a failure. In this connection it must be appreciated:—

(i) The stress of war necessitated embarking on dispersed

mass production without building a prototype and without completing the organization before starting work.

(ii) A large number of the workmen engaged had no experience with submarine work and did not appreciate the high standard of workmanship required. Inaccurate finish led to delay and compromise at the assembly yards. Sections did not always fit and some "pushing and pulling" was often necessary.

(iii) The ambitious target dates set to each stage of the work and the natural urge of the yards to complete accordingly led to a certain falling off of standard, but despite this the yards outstripped the design agency in some cases. As a result such equipment as the telemotor system was never satisfactory, and some of the early hull sections left the second stage yards uncompleted.

(iv) The effect of Allied bombing was felt in varying stages throughout the program.

Despite these factors the first Type XXI U-boat was completed in approximately one year, and within 18 months of receipt of the guidance plans some eighty ships had been delivered and 120 within two years. One assembly yard was, at one stage, producing eleven submarines per month. This can only be described, bearing all things in mind, as a magnificent effort.

It is suggested that a fair conclusion is that the prefabricated scheme for submarines to the extent adopted by the Germans is practicable and desirable provided that:—

(*a*) All design details are settled before beginning mass production.

(*b*) Numbers required are large.

(*c*) Transport means are available.

All of which were fulfilled in the case of the Type XXI and XXIII, except for (*a*). It was, of course, intended that condition (*a*) should be realized, but it was, perhaps inevitably, found impossible to achieve.

In 1941 when the U.570 was captured intact, there was little evidence in this submarine to indicate that the Germans were short of material.

The surrendered U-boats examined in 1945, however, were very different. Nonferrous metals had been almost entirely eliminated and apart from the rather austere appearance inside

the submarines, the amount of maintenance to keep valves, etc., operable due to steel surfaces rusting and the like was very formidable.

Midget Submarines.

In addition to the types mentioned above the Germans devoted no small effort to midget submarine projects. Their interest in this type of weapon was given impetus by the success of the British X craft attack on the *Tirpitz*. These designs were, in some cases, given type numbers as well as code names. In view of the number of projects investigated, reference is made only to those produced in any quantity.

The Biber.—This was a one-man submarine armed with two torpedoes and having a displacement of 6¼ tons. It was fitted with a petrol engine—very clearly an undesirable feature.

The craft is claimed to have achieved a speed of approximately 6½ knots. There were no compensating tanks as such, it being necessary for the operator to adjust the water in the diving tanks for ballast corrections.

The fact that the crew was one man only was found to be a great disadvantage. This was also true of the *Molch*. The principal features of the *Biber* were:—

Displacement	6¼ tons
Length overall	28.5 ft.
Beam (with torpedoes)	4.75 ft.
Diving depth	30 meters
Endurance—	
Surface	125 miles at 5 knots
Submerged	10 miles at 5 knots

324 of this type were built.

The Molch.—This, too, was a one-man submarine designed by a torpedo testing establishment in Germany.

It had a displacement of 10¾ tons and was armed with two torpedoes which were slung under the hull.

This craft was intended to operate submerged only and had a speed of 5 knots.

Its chief advantage over the *Biber* was its ease of production. The torpedo establishment had used as much available equipment as possible and produced practically an entirely cylindrical hull.

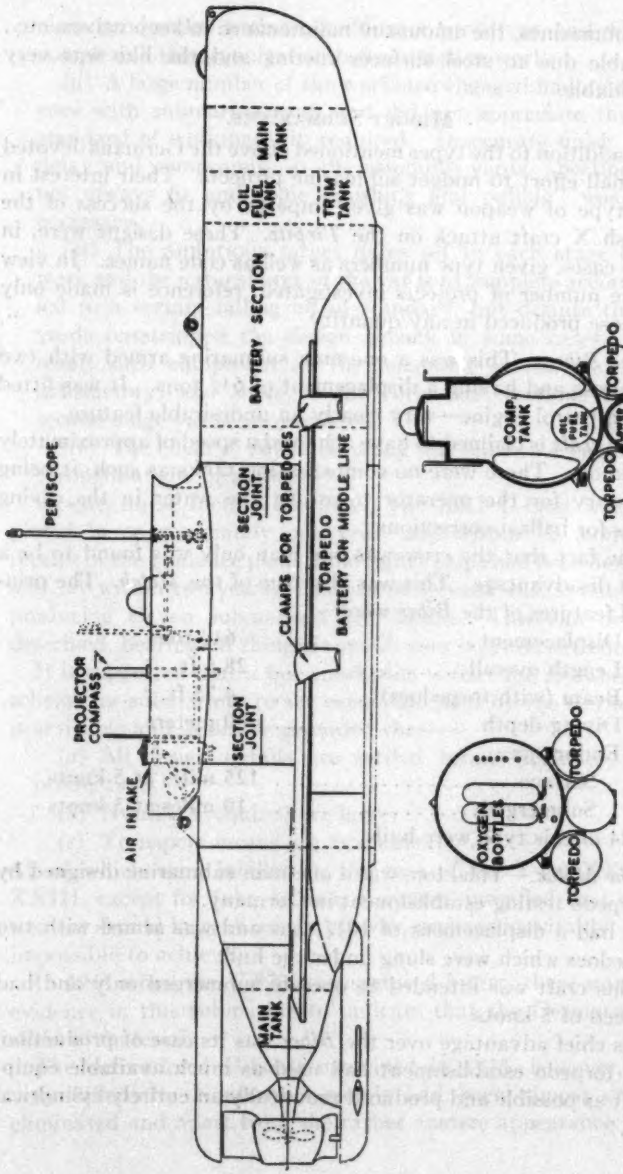


FIG. 4.—OUTLINE OF MIDGET "SEEHUND"

It was propelled by motor and batteries. Its principal features were:—

Displacement	10¾ tons
Length	35.4 ft.
Beam	3.6 ft.
Diving depth	40 meters
Submerged endurance	40 miles at 5 knots

390 of these were built.

The Seehund.—This was probably the most successful of the German midgets and its design was embarked upon after lessons had been learned from a large number of other projects. It was a two-man submarine, and was equipped with a Diesel engine.

Fig. 4 gives a general outline of the design, its main features being:—

Displacement: 15 tons (submerged with torpedoes).
Length: 40 ft.
Beam: 5.3 ft. (with torpedoes).
Diving depth: 50 meters.
Armament—Two torpedoes suspended outside the hull.
Fitted with 60 Hp. Diesel for surface propulsion.
Surface endurance: 250 miles at 5 knots.
Submerged endurance: 60 miles at 3 knots.

250 of this type were built.

The Germans in their work on midgets did not, in all cases, keep the main object in view, i.e., the strict limitation in size. They had hoped apparently that it was possible to achieve a design of midget submarine possessing qualities comparable with larger types. Thus some of the projects, which did not come to fruition, became, in effect, small coastal submarine designs.

One well-learned lesson arising from this was that a midget submarine should be designed for a specific type of operation. An all-purpose midget boat is impracticable.

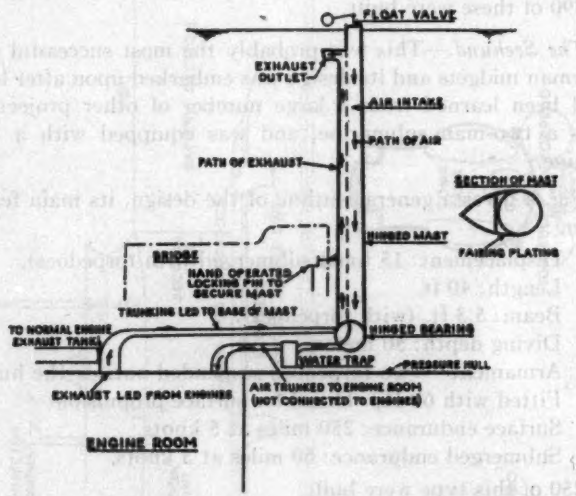
In addition, in some cases, insufficient attention was paid to means of transport of the midget submarines that were produced.

The numbers of the three types produced is one more indication of the effort accorded to submarine projects.

The Schnorkel.

The first U-boats to be equipped with the *schnorkel* were Types VII C and IX. These submarines were already in service so that the new equipment had to be fitted as a modification.

A diagrammatic sketch of the original arrangement is given below.



In the original equipment the *mast* was of the hinged type and contained both the air intake and exhaust trunking. The two leads were taken to the hinge of the mast where they were separated into two runs to the engine-room.

The air intake was fitted with a float valve at the top to prevent the influx of water when "dipping" in rough weather. The valve was shut by the action of the buoyancy of the float.

The air was discharged into the submarine, having been passed through a water trap. It was possible to circulate a good proportion of the fresh air throughout the submarine as the main ventilating fans took their supply from the vicinity of the air inlet.

The exhaust was led from the normal engine exhaust hull valves to the base of the mast and then up to just below the waterline where the trunking was turned aft.

The requirements for the Type XXI and XXIII designs included *schnorkel* equipment, and as the hinged type was resistful and submerged speed was one of the main features of these types, a change was made from the hinged type of mast to a periscope type. The *schnorkel* mast was fitted astern of the periscope, thus avoiding interference with periscope observation which occurred with previous arrangements. The principle of operation was the same as before.

Diving Depth.

Much has been heard of the deep diving capabilities of German submarines. Members of the Allied anti-submarine forces and survivors from sunken U-boats have reported, on many occasions, that U-boats have been to great depths without collapse, figures of the order of 900 ft. being quoted. There is evidence from instruments to support these statements.

Nevertheless, the Germans' own design figures are less than 400 ft., indicating that hull valves, fittings, and tanks were designed and tested to conform to this figure. It may be stated that no German submarine constructed for operations during the last war had a pressure hull capable of a designed operational depth in excess of 500 ft. This statement is made after examination of, and trials in, actual German submarines. It does not suggest that U-boats did not dive below 500 ft., but at the great depths quoted from operations with and against U-boats, it is not exaggerating to say that the margin of safety was very small indeed. The U-boat was not taken very deep from choice and the submarines were not designed to go there as a routine operation.

The Germans did not possess any special quality steel to produce good diving depth and the designs which varied from the circular section pressure hull are not considered to achieve improved performance in this respect, rather the reverse.

Accommodation and Habitability.

Without exception living conditions in German submarines were worse than in Allied submarines, in some cases very much so. This point is probably of more importance in submarines than in any other major war vessel, space being at such a premium.

A comparatively small number of bunks was provided, far less than the aim of one per man in our own submarines. In addition, the number of the crew in the U-boats was less than other navies would have allowed, giving rise to greater strain on the users.

The messing, washing, and sanitary arrangements can only be described as inadequate. It was, in general, only possible for some 20 per cent of the ship's company to be seated for meals at one time. Of course, it may be argued that the Allies set too high a standard, but a few days at sea in any submarine would probably dispose of this view.

The Germans themselves did, in fact, come to realize the error of poor accommodation, because the Type XXI saw a definite improvement, though still below Allied standard, and the fitting for the first time of air conditioning equipment.

Conclusion.

It is felt that the most outstanding feature of the story of the German U-boats of the 1939-45 War is the numbers produced. Beginning with a fleet of some 57 boats, the Germans were to produce numbers, of the most varied types, equivalent to *a new submarine every second day for the whole duration of the War*, this at a time when the nation was engaged in major warfare on the land and air fronts and subjected to most severe air bombardment.

No paper such as this can give a complete picture of the work done by the Germans or the range of developments explored. To cover the whole story would take a number of volumes.

The Type XXI was not tried in war, but few will say that it would have been defeated at once or that it was the last word in submarine design.

Thus, then, to a country such as ours, absolutely dependent on the passage of ships on the sea, the story of the U-boat effort should be a grim warning. The submarine is still far from defeated, and to combat a pro-submarine effort on similar scale would require an even greater anti-submarine effort.

MARINE POWER PLANTS OF THE FUTURE.

The following article by F. V. Smith is reprinted from "Marine News" for May 1948.

What the maritime industry needs at the present time is a competent mystic to guide it through the maze of newly created technological developments. Never before in the history of engineering have the design and application engineers become so submerged in a sea of new tools and new ideas. Before these can be fully digested and molded into workable patterns a new batch of ideas invariably arrive from the research and engineering laboratories to further confuse the issue. One would be rash, indeed, to predict any limitations for the future. All we know at present is that we are on the road from here to there, destination unknown, without traveling on any definite time schedule.

In the already established fields of power plant engineering the frontiers are gradually being pushed ahead by the old familiar pioneer processes. Scanning the Central Station power field from which the marine power plant designer draws many of his inspirations it is noted that the million Kw. mark has been passed for power plants operating at steam temperatures of from 900 F. to 1000 F. and that six turbine-generator units having a total capacity of 700,000 Kw. are in course of construction which will operate at a steam temperature of 1050 F. Three of the latter units are rated 125,000 Kw. each and will operate at a steam pressure of 2000 lbs.

With our present knowledge of materials, temperature is more of a limiting feature than pressure. The laws governing high temperature material performance are gradually becoming better understood. Rigorous testing procedures for metals have been evolved for such items as behavior under vibration, heat shock, stress, temperature variation, corrosion, ductility and creep.

Continuous research is being made in relation to fuels for special purposes and some day the findings may be of commercial value. Atomic energy also offers many possibilities.

It is unsafe to predict too far ahead into the future or to close the door to any new line of adventure by saying that it can't be done. To the scientist all things appear to fall within the realm of possibility. The engineering economist, on the other hand, is apt to view things through dark glasses and delve into fields of

probability rather than fields of possibility. These conflicting viewpoints of the scientist and economist may best be understood by referring to a little past history.

Inertia to Change.

A little philosophy mixed with good hindsight and a dash of economics should quickly dispel one of the idea that a new Utopia in engineering lies just around the corner which will be subject to immediate and wide application. There is a lot of inertia against sudden and sweeping changes which tend to upset well intrenched social and economic orders.

The inertia against the discarding of the old, and the adoption of the new is no more self evident than it is in the marine field.

Reciprocating engines, hand fired Scotch boilers and other engineering apparatus in vogue at the turn of the century are still in evidence, irrespective of the fact that turbines and Diesel engines have been highly developed for a period of two decades. In the year 1946 fifty-three per cent of the ships in the world were still powered by reciprocating engines, and over one-third outfitted with Scotch boilers.

The fact that the sum total of all ships in the world consists of all age groups in itself becomes a barrier against sudden change. In the past there has been a gradual lopping off of the old at the bottom and a replacement with the new at the top. Between these two age groups are found all gradations of machinery types from the primitive to the modern.

On the Great Lakes, where a large portion of our merchant fleet sees service, the majority of ships are still powered with reciprocating engines, hand fired Scotch boilers and jet condensers. A similar condition exists on our inland waterways, in our harbors and in our coastwise shipping lanes.

Sitting back and waiting for things to die of old age before replacing with the new is a slow evolutionary process.

Economic Considerations.

It is a nice thing to start on a clean sheet where one can discuss the pros and cons of modern engineering without first having to clear the deck of a litter left by previous generations. This method of attack does not come often nor does it offer a success-

ful modernization program which will make sweeping changes in all age groups. As to the immediate need for such a program, however, there can be little doubt, as all uneconomic forms of power generating apparatus are becoming a serious drain on the fuel resources of our country.

Comparison of the fuel rates on reciprocating engine driven ships with any of the modern forms of propulsion apparatus show a differential as high as fifty per cent in favor of the new. As fuel prices rise the position of all antiquated types of powering equipment becomes more untenable.

Speed is also a very important factor in the evaluation of ship economics. Modern powering apparatus permits the application of approximately twice the amount of propulsive power as the antiquated types for the same expenditure of fuel; the same amount of power for one half the amount of fuel; or, any gradation of values between the two extremes. Increasing the speed of a ship increases its potential earning capacity and permits a fewer number of ships to perform the same service. A reduction in fuel consumption without speed change increases the potential earning capacity of a ship only insofar as advantage can be taken of the differences in fuel weights.

A solution of the economic problem involved in a modernization program effecting ships in the middle age group is much more difficult of solution than the economic problems relating to new ships. The tendency in the past has been to hang on to any apparatus which is bought and paid for as long as it will work. Even fifty per cent fuel savings have failed to make a dent into this knotty old problem.

In the design of a new ship the goal sought is maximum overall economy, or, stated in another way, the minimum cost per ton mile for transporting freight. How this is attained is of little importance. All factors such as power, speed, dead weight tonnage, first cost and operating costs can be placed on a sliding scale and a series of calculations made to determine optimum values.

In considering the power plant on a ship, which is but part of the overall economic problem, there are several forms of propulsion apparatus to choose from, and also overlapping fields of application for each type.

Today one can make an unbiased analysis, arrive at definite conclusions regarding the most economic form of propulsion apparatus to use and then be all wrong in his predictions five years hence. Fuel costs alone may change the economic picture over night.

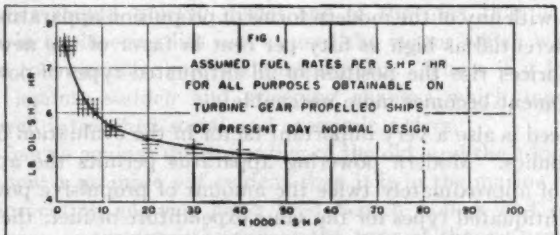


TABLE SHOWING MONETARY SAVINGS FOR ONE PER CENT SAVING IN FUEL CONSUMPTION BASED ON 300 DAYS OPERATION PER YEAR AND FUEL AT $14.00 PER TON

S.H.P.	2000	6000	8500	12500	20000	50000	100000
FUEL RATE	.75	.63	.60	.57	.54	.5	.5
TONS OIL YR.	4821	12150	16393	22902	34714	80357	160714
1% SAVING	482	1215	1639	229	347.1	803.6	1607.1
VALUE	$675	$1701	$2295	$3206	$4859	$11250	$22500

F.V.S. 3/3/48

Evaluation of Fuel Economics.

In meeting the demands of an industry for more economic forms of power generating equipment the relationship between first cost, and fuel costs plays an important part. The relationship between these two factors can only be correctly evaluated if plotted on a scale of variable power.

In the low power field of application in which the annual fuel consumption is also low, costly capital expenditures are rarely justified for the attainment of small percentage fuel savings. Such expenditures are, however, almost invariably justified in the high power field of application.

The relationship between power rating and fuel savings is clearly shown in Fig. 1 and the table accompanying it. The graph of fuel rates is approximately correct for turbine-gear propelled ships of present-day normal design burning the cheaper grades of bunker oil. The table showing the monetary change for a one per cent difference in overall efficiency is based on 300 days operation at sea and a fuel value of $14.00 per long ton.

In referring to the table it will be noted that a one per cent differential in fuel amounts to 48.1 tons per season for a 2000 Shp. installation, 121.5 tons for a 6000 Shp. installation and 1605 tons for a 100,000 Shp. installation. If the additional capital expenditure required to secure this one per cent gain is amortized over a ten years' period, a capital expenditure of $6,750, $17,010, and $225,000 would be justified in the three respective cases each per cent reduction in fuel consumption.

Today the designer has an almost unlimited number of variable factors to choose from. In the case of a steam vessel, the initial steam pressures and temperatures offer a wide range for choice. Likewise, there are binary cycles, regenerative cycles and heat cycles all waiting on the shelf ready for use if need be. Proper coordination of the design factors in most instances pays the best dividends.

A few conclusions may be drawn from the abbreviated table of fuel savings versus justified capital expenditure. The first of these is the growing need for standardized marine propulsion power plants in the low power field which will eliminate the high cost of making special designs for each individual application. There is a decided trend in this direction at the present time and it may be safely predicted that in the not too distant future ship owners will be able to purchase power plants in standardized ratings the same as they now purchase small motors. This trend towards standardization is in keeping with the type of manufacturing facilities which have been developed in this country.

The fight for lower first cost on the one hand and lower fuel costs on the other offers plenty of opportunity for a clash of ideas between the design engineers with their idealisms, and the engineering economists who hold the purse strings.

Overlapping Fields of Application.

At the present time there are many overlapping fields of application for the existing types of propulsion power plants if considered from the standpoint of power rating alone. The choice, however, narrows down to a single type for an individual ship plying a given trade route if such choice is made without bias and strictly in accordance with the findings of a complete economic analysis.

The only trouble with an analysis of this type is that it reflects conditions as they exist today and not the conditions that may exist ten years hence or within the life of the ship. In the march of time, all things built today will become the antiques of tomorrow.

Viewing the subject in the light of present-day economic values all types of propulsive power apparatus overlap in the lower power field of application. The number of competitive types lessens as the power per propeller shaft increases, leaving only turbine gear drive and turbine electric drive in the upper brackets.

In the low power field of application, that is from a few hundred horsepower up to 4000 horsepower per shaft Diesel electric drive has a lot to commend it. Such drives can take advantage of a multiple number of high-speed prime movers. The variable speed reduction characteristics, and the excellent maneuvering characteristics of direct-current generators and motors cannot be excelled by any other type of powering apparatus. This type of drive is applicable to a wide variety of ships, and has made greater progress in standardization than any other. Because of the similarity of railroad locomotive applications, the progressive refinements made in this field become readily available to the marine field.

Diesel electric drive using alternating current generators and motors has a higher power range than the direct current system, but lacks the variable speed reduction characteristics of the d-c system. All propeller speed variations are made by adjusting the speed of the prime movers. The crash reversal torque characteristics also leave a lot to be desired and up to the present time dynamic braking has been used to first slow down the ship before propeller reversal.

Turbine electric drive can be designed for any power output desired within the limitations of propeller design. Multiple generator units can be installed and switching arrangements provided for operating a multiple number of shafts from a single unit at reduced powers. It is versatile from an installation point of view and in some types of ships advantage can be taken of short line shafting, thus offsetting to an appreciable degree its primary weight disadvantage. Electric drive ships have in many instances been used as floating power plants to supply

power for shore use. As with other forms of drive, its application must be justified by its overall economic advantages.

Turbine-gear drive encompasses by far the largest field of application. It now holds a weight, efficiency and first cost advantage, and like turbine electric drive is adaptable in all powers within propeller design limitations.

Diesel engines when coupled directly to the propeller shaft are limited in their rated power output and are also heavier than turbine gear drives. Their economic field of application depends upon the differential in fuel costs at ports of call, a true evaluation of lubricating oil costs, and the extent to which the lesser weight of fuel carried can be credited to additional earning capacity.

Reciprocating steam engines are heavier than the modern forms of drive and because of their unequal crank effort require larger diameter line shafting. Their efficiency is low and on an average they consume roughly 100 per cent more fuel than modern steam drives. The one strong point in favor of the reciprocating engine is its fine maneuvering torque characteristics which is only excelled by the Diesel electric drive system nsiug direct-current generators and motors.

Probable Effect of Fuel Costs in the Future.

Fuel is truly the foundation on which our present day civilization rests. Without coal and oil we would quickly revert to the primitive. At the present time over 2000 products are obtained from the hydro-carbons contained in oil and coal.

The drain on our fuel reserves is becoming more serious every year. The increasing population ever increasing number of automobiles, greater use of oil for home heating, and the change over to Diesel locomotives by many of our principal railroads all point in one direction—that of increasing prices rather than decreasing.

All avenues for obtaining fuel oil in the future are being investigated. These include the possiblity of obtaining fuel oils and gasoline from natural gas, from coal and from underground gasification of coal in the mines. Other lines of research pertain to the obtaining of oil from tarry sands and by the gasification of celluose products such as cornstalks, sawdust, and pulp residue from paper mills.

The studies now being made regarding the use of atomic energy and the creation of new types of fuels are but an extension of the idea of finding additional forms of energy which can be converted into power, light and heat.

By force of necessity we may sometime in the future be compelled to adopt the most economical forms of power generating equipment whether we wish to or not. Fuel may even be rationed for essential services.

It is obvious that any economic analysis made today based on present fuel prices may be entirely out of line for future predictions.

Avenues for Future Improvement.

1. *Steam Turbine Design*—The development of new materials suitable for use in high pressure, high temperature steam power plants has been a subject of constant investigation and research over a long period of years. Such research is being stepped up at present, and long range programs are projected for the future. During the past twenty-five years steam pressures and temperatures have been gradually increased in the Central Station field from 350 lb. 650 F. to 2000 lb. 1050 F. This pioneer work is available to the marine field any time that such application can be economically justified.

The physical and mathematical laws pertaining to heat and energy have been the subject of investigation for years and such studies are still being continued. The laws now established far transcend present day fields of practical application.

Available energy diagrams and formulas have been extended to 5500 lb. pressure at 1600 F. for steam, and similar diagrams are being prepared covering air and the gases which may find future application in gas turbine design.

The thermodynamic gains to be realized through the incorporation of the reheat cycle, regenerative cycle and binary cycle have also been calculated for a range of values extending far beyond present day fields of application.

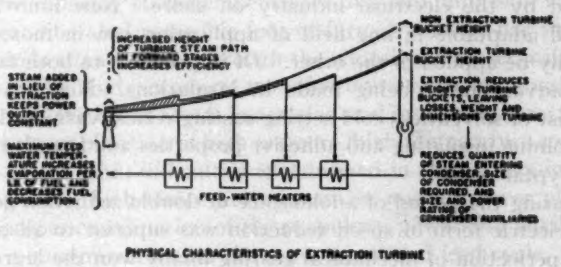
A thorough investigation is also being made of turbine speeds with the object in view of further reduction in weight and dimensions. In the high-power field of marine application, turbine gear units have already been built with prime mover

speeds ranging from 6000 to 7500 Rpm. Turbine rotor speeds of 10,000 Rpm. have become quite commonplace in the auxiliary turbine generator field of application.

Still higher turbine speeds can be safely predicted for the future.

Turbines as now designed for marine applications generally perform a dual service, that of furnishing propulsive power, and that of furnishing extraction steam at various pressure levels for heating the feed water and other miscellaneous heating purposes not strictly related to the propulsive power cycle.

Steam extraction has a very beneficial effect on the turbine design, efficiency, and overall cycle efficiency of a steam power plant as shown in Fig. 2.


NON EXTRACTION TURBINE BUCKET HEIGHT
INCREASED HEIGHT OF TURBINE STEAM PATH IN FORWARD STAGES INCREASES EFFICIENCY
EXTRACTION TURBINE BUCKET HEIGHT
STEAM ADDED IN LIEU OF EXTRACTION KEEPS POWER OUTPUT CONSTANT
EXTRACTION REDUCES HEIGHT OF TURBINE BUCKETS, LEAVING LOSSES, WEIGHT AND DIMENSIONS OF TURBINE
MAXIMUM FEED WATER TEMPERATURE INCREASES EVAPORATION PER LB OF FUEL AND DECREASES FUEL CONSUMPTION.
REDUCES QUANTITY OF STEAM ENTERING CONDENSER, SIZE OF CONDENSER REQUIRED, AND SIZE AND POWER RATING OF CONDENSER AUXILIARIES
FEED-WATER HEATERS

PHYSICAL CHARACTERISTICS OF EXTRACTION TURBINE

Figure 2

2. *Reduction Gear Design*—Mechanical gearing of the double reduction herringbone type have reached a high state of perfection and are available in all powers up to the limitations imposed by propeller designs. Such gears are practically noiseless in operation and have excellent performance record. It has taken twenty-five years of difficult plodding to attain this goal.

Taking a long range view into the future, it is difficult to picture coming generations of engineers sitting idly by accepting present day designs with a laissez faire attitude. It seems more than likely that they will try to pave new roads of their own regardless of how rocky the path may be. A little of this pioneer spirit is already showing up. Spurred on by the planetary gear developments in the small power field, engineers are now speculating on the possibility of extending the application of this type

of gearing to the high power field. If such an extension could be made, then another radical change might be brought about.

Because of the compactness and light weight of planetary gears they might be used in a number of different ways. That is, in both the straight mechanical forms of speed reduction, and also in connection with turbine electric drive. In the latter form of drive a planetary gear could be placed between the turbine and generator, thus freeing the turbine from any speed limitations. A similar gear could also be placed between the motor and propeller shaft thereby greatly reducing the number of poles on the motor and its diameter.

3. *Turbine-electric drive*—Turbine electric drive up to the present time has followed more or less orthodox practices established by the electrical industry on shore. New innovations found adaptable in one field of application can in most cases readily be applied in the other. Of importance to both fields is the advancement being made in insulations. New materials consist of solventless cold setting casting resins, varnished cloth combining insulating and adhesive properties and improvements in glyptal resins.

During the period of adolescence of double reduction gearing the electric form of speed reduction was superior to all others. The perfection of mechanical gearing finally won the laurels for turbine gear drive as regards weight, space, and efficiency. Gaged by present day standards this handicap is not sufficiently great to warrant the prediction that turbine-electric drive is permanently out of the race. It is not wise to close the door on any given type of propulsion equipment as one can never tell when some new innovation will turn the tables.

4. *Diesel electric drive*—At the present time the power limitations of Disel-electric d-c drive is based on the power and speed limitations of the prime movers. This is approximately 2000 horsepower at 850 Rpm. Coupling three such units in series gives an output power of slightly over 5000 horsepower at the propeller shaft.

Both of these factors—power and speed—should be viewed with an open mind as they are based on horizontal shaft designs. Suppose the prime movers were built with vertical shafts and the cylinders arranged around a circle, what then would be the

limitations? Additional layers of cylinders could be added pancake fashion to get any power desired. When commutation becomes a limiting feature with direct-current, a switch could be made to alternating-current, where no such limitation applies.

Looking towards the future of Diesel electric drive it appears that the relative fuel costs will play a greater part in dictating its application than its power limitations.

5. *Future Advancements in Boiler Design*—The chemistry of combustion places definite limitations on boiler efficiency. Modern oil burner and automatic combustion control equipment now provide a means for obtaining maximum efficiency with minimum safe margin of excess air. Practically all modern boilers are provided with either air preheaters or economizers or both, for reducing stack temperatures to a minimum safe value. The heat liberated per cubic foot of furnace volume has shown continuous advancement through the adoption of water walls.

The boiler and accessory manufacturers have kept well in advance in the art of steam generation and in no case have they limited the application of high pressure, high temperature steam.

One of the main difficulties encountered in the application of high-pressure, high-temperature steam in marine power plants is the proper coordination of the propulsive power and auxiliary power requirements within the same steam and feed cycle.

In a high pressure power plant—850 lb. 900 F. and above—it would be much better if these two problems could be divorced so that each service stood on its own feet. This would also simplify the change over to port operation. A secondary boiler plant furnishing lower pressure steam for heating and other miscellaneous services would eliminate the many high pressure reducing valves with their attendant throttling losses. This boiler could be of the separately fired type, or make use of a liquid type of heat transfer agent from the main boilers as the heating medium.

Present Aspects of Speculative Power Plant Developments.

Research and developmental engineers do not work on a time schedule. It is their job to take an idea in the raw, then laboriously try to develop a preliminary working model built

on the most elemental lines. Among a few such adventures in this speculative field of engineering today are such items as high speed turbines operating at speeds above 20,000 Rpm., high speed motors, motors to operate at 500 F., gas turbines and atomic power plants. No one knows when these preliminary models will be successfully developed, nor how long it will take for the finishing touches to be added which will make them commercially feasible.

1. *Gas Turbines*—The proper way to look upon the gas turbine developments now taking place is that a new road is being paved, destination unknown. The future of gas turbines looks bright if viewed from the number of its favorable attributes which can be put down on paper. These consist of the substitution of a packaged power plant consisting of an air compressor, turbine, and fuel oil burner, for a steam power plant consisting of a boiler, turbine, condenser and a multitude of auxiliaries. The elemental power plants now in course of construction by a number of the leading manufacturers are the proving ground for new heat resisting metals rather than fuel efficiency. The metallurgical problems are the most important at the present time requiring solution. After satisfactory solution of this problem, regenerative cycles, intercooling of the air in the compressor and other items tending to increase the cycle efficiency will be added.

Elaborate testing equipment is being used to study all phases of the operation of the elementary power plants now being built, and millions of dollars have been appropriated for the future development and refinement of this type of power plant.

2. *Atomic Power Plant Development*—This paper would not be complete without a few words regarding the future of atomic power plants.

Civilization has made great progress in the material arts during the past five centuries by applying the laws of chemistry and physics which are both based on the premise that matter and energy can neither be created nor destroyed. We are now entering a new era in which a third law has become fairly well established, and that is that matter and energy are interchangeable.

In the well known laws of combustion governing fuels a definite amount of heat is liberated per pound of combustible oxidized. The products of combustion can be disassociated

chemically, returned to their original state and weighed without showing any perceptible loss of weight in the process.

In the conversion of matter into energy, the energy is equal to the mass multiplied by the square of the velocity of light. The astronomical results shown by such calculations are tremendous even when discounted 99.9 per cent.

Fictional writers already have dreamed of ships which can travel several times around the globe on one pound of fissionable material, trips to the moon on rocket ships, and the operation of all central station power plants in the United States for a year on 40 pounds of such materials. It is not the purpose of this paper to discount the possibilities of the future, but at the present time it appears that we are a long ways off from this Utopia.

From what one can glean from the official reports and published treatises on the subject of atomic energy it appears that the chain reactions in uranium can now be controlled from a slow graduated, to an explosive degree with certainty. The problem that now confronts the scientists is how to dispose of the products of disintegration which are lethal in effect.

Another problem exists and that is finding a suitable and non-contaminable heat transfer agent to carry the heat from a uranium pile to a steam generator, or to an air or gas stream for use in a gas turbine.

A number of metallurgical problems closely related to both of the foregoing factors also await solution.

One thing is certain and that is that atomic power plants will not be marketed until ways and means are found for fully safeguarding everyone connected with their operation.

We Have Reached the Questioning Age.

In the field of research one questions everything, especially standardized practices. All variable factors are placed on a sliding scale and projected from the known fields of application into the unknown.

If one take a broad view of a simple thing like the established frequency of 60 cycle current, which has become fairly well established in this country, he might in all fairness to his freedom of thought question its use. The use of 60 cycle frequency limits

the top speed of a motor to 3600 Rpm. If frequency is placed on a sliding scale, 100 cycles would give a top motor speed of 6000 Rpm.; 200 cycles, 12,000 Rpm.; 400 cycles 24,000 Rpm., and etc.

By plotting motor weights, dimensions and costs against cycles, the optimum value of frequency might be found to be something entirely different than 60 cycles.

A few high speed motors have been built; a 22 Hp. motor operating at a speed of only 12,000 Rpm. had a diameter of 4½″ and a length of 12″; a 100 Hp. motor operating at 10,000 Rpm. had a diameter of 8″ and a length of 16″; and a 4 Hp. motor operating at 18,000 Rpm. a diameter of 2″ and a length of 7⅝″. Refrigerants also might conceivably be used as coolants.

Another item that might come under the scrutiny of the developmental engineer is the matter of voltage. Why have we standardized on 110,220, and 440 volts for auxiliary motor drives?

The temperature limitations placed on electrical equipment have been arbitrarily set at given values based on the types of insulating material available at the time the restrictions were made. New types of insulating material have been developed which permit motors to operate successfully at temperatures up to 500 F. Can advantage be taken of such a new development, is a perfectly legitimate question.

Has full use been made of the new electronic types of control? It is now possible to use small differences in temperature, pressure, light intensity, sound, color, stress, current, and voltage to initiate electronic changes which can in turn be amplified to any value desired to effect control change. Have such devices any practical use aboard a ship, and if so, how are they to be applied?

A number of dead end streets have been reached in conventionalized designs, and these are detrimental from the standpoint of progress. What is needed at the present time to overcome these hurdles is creative imagination.

In the long process of working out new developments each step of the journey unfolds new problems to be solved. Many times the by-products of these long investigations prove to be of greater value than the success of the original venture.

The research laboratories are full of such fruits of research.

CONCLUSION.

The young engineers starting out today have a long, interesting and exciting road ahead of them. The old engineers who are now winding up their careers were born thirty years too soon. Present generations may laugh at the crude inventions now reposing in museums, but let us all hope that future generations may also have the same laugh at our expense. This is what we call progress.

PRESENT POSITION OF STEAM.

The following is an editorial appearing in "The Marine Engineer," Annual Steam Number for 1948.

With the appearance of our second post-war additional Annual Steam Number nearly three years after the conclusion of the Second World War, it is a good time to take stock of the position which steam holds for the propulsion of merchant shipping. During the war, ships were fitted with whatever type of machinery could be most easily supplied. Demands for war production placed a limit on ship machinery types, and the best use of what production facilities were available sometimes meant that steamships were built for owners who under normal conditions would have preferred to take delivery of motorships as being more suitable for their particular trade. A degree of austerity was necessarily introduced into provisions of both main and auxiliary machinery as with the hull itself. Cargo steamships were for the most part fitted with machinery of severely standardized type designed for quantity production, in which economy of operation took second place to ease and speed of production. Nevertheless it is somewhat remarkable that British shipbuilding and marine engineering were maintained at such a high level of workmanship and finish during the war years. Towards the end of the war some very fine cargo liners were built, but even so, they were not always capable of that speed in service which was desirable for peace-time operation.

Sufficient time has elapsed since those gruelling days for owners to order ships built to their own specification, and it is instructive to see the trend of steamship tonnage output and steamship machinery choice as shown by the latest figures supplied by Lloyd's Register. The tonnage of vessels fitted with internal-combustion engines launched in Great Britain and Northern Ireland in 1947 was 47.2 per cent of the total output of steam and motor tonnage during the year. The figure compares with 47.3 per cent for 1946 and 45.4 per cent for 1938. As between 1946 and 1947 therefore the steam-motor ratio is almost constant and it may well be that the steady increase in the proportion of motorship to steamship construction evident before the war has come to an end. Meanwhile it is to be noted that of the 16 tankers of 1000 tons or over launched during the year, only five were steamships. In the United States, of the total of 164,848 tons launched last year, 125,659 tons (76.5 per cent) consisted of steamships, all fitted for burning oil fuel. All these vessels, with the exception of one of 953 tons, are turbine-driven. It is perhaps not surprising that in Denmark the tonnage launched—60,695 tons—was composed almost entirely of motorships, but in France 16 steamships totalling 28,430 tons were launched representing 31 per cent of the total, which shows that steam is playing a useful role in that country for the propulsion of quite small vessels. All except two were of straightforward steam reciprocating type.

Total steamships launched in the world in 1947 consisted of 257 vessels of 934,037 tons, of which 850,000 tons (91 per cent) were fitted for burning oil fuel in their furnaces. Motorship tonnage launched was rather greater at 1,168,584 tons. It is interesting to see just how this steam tonnage is divided between the different machinery types. One hundred and seventy vessels of 302,173 tons were to be fitted with steam reciprocating engines, 25 steamships of 95,589 tons with a combination of reciprocating and turbine engines, 61 vessels of 524,075 tons with geared turbines and one (a tanker) of 12,200 tons with turbo-electric drive. The machinery types covering tonnage launched in Great Britain and Northern Ireland during 1947, apart from the turbo-electric tanker *Hyalina*, were divided between the different main types as follows:

Reciprocating engines		*Combination machinery*		*Geared turbines*	
No.	*Tons*	*No.*	*Tons*	*No.*	*Tons*
106	197,734	14	59,639	35	360,490

There is one further factor to be observed regarding British launchings last year. Steam tonnage built for burning oil fuel was about 590,000 tons, leaving approximately only 40,000 tons constructed for coal burning, a state of affairs which is hardly surprising in view of the scarcity of bunker coal and its excessive cost.

The foregoing shows the present position of steam propulsion in some detail. It is the aim of this specialized number to give in supplement technical particulars of a few of the interesting steamships recently completed or now fitting out. Emphasis, it will be noted, is on turbine machinery and water-tube boilers; but we have shown that the reciprocating steam engine and combination machinery usually in association with Scotch boilers are still popular, as last year's Steam Number brought out. More advanced steam conditions—and hence higher economy—are the trend of the moment; and had this issue been published a little later it could have recorded the completion of the first of several high-powered single-screw British geared turbine vessels with somewhat higher steam pressures and greater superheats than the "450-750" state which has served so well for several years. Steam propulsion is about to take an important technical forward step.

NONMETALLIC BEARINGS.

I—Some Characteristics of Soft Rubber as a Material for Water-Lubricated Bearings.

By A. Bednar.

The following articles, from a symposium contributed by the Rubber and Plastics Division of the A. S. M. E., are reprinted from "Mechanical Engineering" for July, 1948.

Many years ago, an engineer, operating mining equipment in the west, presumably far removed from the sources of supply, was having much difficulty in maintaining bearings in a pump required to handle muddy and sandy water. The sand and grit would cut out the bearings at a rate far exceeding the rate of supply, and so inevitably he eventually ran out of spares.

In desperation, the story goes, he wrapped a section of discarded rubber hose around the shaft in the place of the bearing, and this arrangement, it is said, not only outlasted the original bearing several times, but caused much less wear to the shaft. That, supposedly, was the first application of soft rubber as a water-lubricated bearing.

Another legend describes the inventor as wearing rubbers on a rainy day, slipping on a wet manhole cover, and being jarred into the idea of water-lubricated rubber bearings; a sudden thought apparently, something like the Eureka episode of ancient Greece.

Be that as it may, these accounts indicate some desirable properties of soft rubber for bearings under conditions of water lubrication. Water lubrication, as applied today, generally because of necessity, imposes an abrasive problem of varying degree in almost all cases.

Twenty Years of Service.

Bearings of soft rubber composition, both natural and synthetic, have now been in use for over 20 years, successfully, under conditions requiring water as a lubricant. These applications include stern-tube and strut bearings of large ships and boats involving shaft diameters from fractions of an inch to more than 2 ft.; bearings in pumps, such as deep-well turbines, circulating

pumps, dredging pumps, those used for drainage and irrigation, and others. Rubber bearings perform almost unbelievably well at the outermost end of the cutter shafts on hydraulic dredges where they support the rotating cutter as it digs into the bottoms of rivers and harbors. Sand washers and classifiers, and agitators used in many industrial processes employ rubber bearings advantageously. Large rubber guide bearings for hydraulic turbines of 30 to 40 in. shaft diam. are being used, some recording continuous service of more than fifteen years' duration.

Careful studies and tests carried out at different times over a period of years by research laboratories of the B. F. Goodrich Company, and observations in service, indicate that these soft-rubber materials, when properly prepared, have the ability to deform and so allow abrasive particles to pass between the resilient surface of the rubber bearing and the revolving shaft with minimum pressure against these surfaces. In other words they do not momentarily carry a large portion of the total bearing load as may be the case with more rigid materials. Moreover, these particles do not embed themselves permanently in a properly compounded soft rubber where they could score the shaft, and consequently abrasive action is to an overwhelming degree nullified. Soft rubber has clearly demonstrated its unique property of wear resistance in many other applications under severe conditions, most common of which of course is the automobile tire.

Experimental Work Reported in 1932.

Experiments conducted some years ago by Dr. W. F. Busse and Mr. W. H. Denton the results of which were published in 1932, demonstrated the behavior of coefficient of friction for soft-rubber bearings under varying ranges of load and speed. Values of less than 0.02 are recorded in these experiments under the conditions of test approximating 300 to 700 Fpm., and up to loads of 45 Psi. of projected bearing area. At these operating speeds and above these ranges, there appears to be no increase in the coefficient of friction with the increase of load. In fact, other experiments made prior to that date, and also since that time, indicate that under some conditions coefficient of friction tends to decrease with increase in load at given speed. Here

again the ability of soft rubber to deform serves an excellent purpose in that the bearing surface may adjust istelf to minute irregularities in the shaft and so permit the comparatively thin film, characteristic of a low-viscosity lubricant like water, to form and be maintained between the moving surfaces.

Furthermore, in a successful bearing design, featuring a number of longitudinal bearing faces around its inside surface supporting the shaft and separated by water grooves, the flexible nature of rubber allows the faces to deform so as to vary the angle of approach between the bearing and the shaft to correspond to the varying conditions of load and speed, producing an effect perhaps

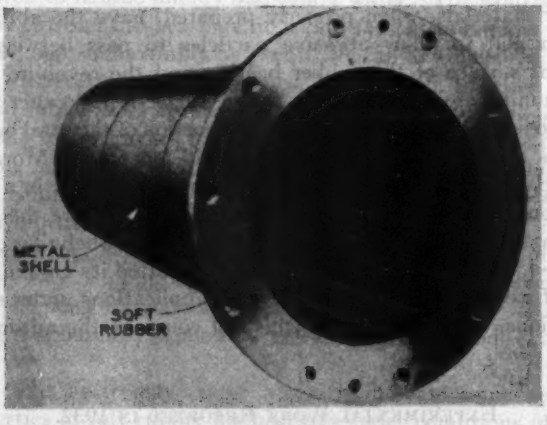


FIGURE 1.—Water-Lubricated Rubber Bearing for Ship's Propeller Shaft, Approximately 15 Inches Diameter.

similar to the action of pivoted segments of modern thrust bearings. This, according to Busse and Denton, probably accounts for the lack of effect of load on the coefficient of friction over the ranges they tested.

The flexible property of rubber provides another advantage on high-speed shafts in that it permits the shaft to turn on its center of giration when it does not coincide strictly with its geometric center, thus reducing vibration and stresses in the bearing and supporting structure.

High-Speed Applications.

The upper limit of speed for water-lubricated rubber bearings has not been determined. Laboratory tests up to approximately 1500 Fpm. have been made, which would cover most of the common operating ranges. However, on a special racing boat, built in England some years ago for speed-record runs and for races here and abroad, rubber bearings were used for a 1⅜-diam. shaft turning at 12,000 Rpm., or approximately 4330 Fpm. These bearings showed absolutely no distress after having served in various test runs, races, and numerous record runs on this boat.

In continuous service for several years are rubber guide bearings on high-head hydraulic turbines in western states involving shafts 15 in. diam. at speeds of more than 500 Rpm., or on the order of 2000 Fpm.

Like other mechanical devices, rubber bearings require appropriate conditions for proper operation. Shaft journals should be smooth, and consequently reasonably hard corrosion-resistant materials are necessary. Certain grades of bronze, properly processed, and monel metal have been found satisfactory in sea water. Stainless steels and even chrome-plated materials are good in fresh water and in many industrial liquids. Metallized shafts have given good service in some applications, but this method of applying noncorrodible, and at the same time good wearing surface, requires well-experienced operators. Generally, for continuous operation, the bearing should be open at both ends for free circulation of water when submerged, or if the location requires partial or total enclosure of the bearings, provision for circulating water through them should be made. On the other hand, for extremely slow speeds or intermittent service such as rudder pintle bearings, no special provision for water circulation is required even if one end of the bearing is closed.

For natural-rubber compounds oil should be avoided, but recent developments of synthetic compounds for bearings have eliminated this precaution. Tests and service experience indicate that synthetic-rubber compounds promise many worthwhile improvements in other fundamental features of the soft-rubber type of bearing, and this may perhaps be more completely discussed in a future paper.

II—Some General Information About Lignum-Vitae Bearings.

By H. V. Townsley.

There is a much more interesting aspect to lignum vitae than that of using it as a bearing material. Its commercial use began about 1520, when it was imported into Europe for medicinal purposes. The fact that the splinters from this material which break through the skin will not suppurate or fester, led doctors to believe that it had great healing properties, and for 200 years thereafter it was the main ingredient for medicines; until recent years it was still used as a cure for certain diseases. Because of these alleged healing characteristics, the wood received its name "wood of life." Even today the gum has a place on the apothecary's shelf since it is extracted from the sawdust and added to lard and fatty canned foods to eliminate rancidity.

Some time after lignum vitae was put to use as a medicine, someone discovered the fact that rope sand cables would slide over this material with less resistance than over other woods or materials. Its first use as a bearing material was in ship construction, and it is still used to a considerable extent in ship parts.

Lignum vitae or Guayacum, is a hard dense wood containing an oily resin. The tree is usually found in the tropics and is most abundant in the Island group of the West Indies. It has a specific gravity of 1.35 and weighs 83 to 85 lb. per cu. ft. From this you can understand why it sinks in water.

Harvesting and Working Lignum Vitae.

The tree grows as large as 3 ft. in diam. and 30 ft. high. Because of its great weight and lack of logging facilities in the West Indies, where most of the commercial lignum vitae comes from, it is usual to harvest the tree from 4 to 20 in. in diam. Most of the bearing material is found in 8 to 12-in.-diam. trees. All logs are cut for transportation into 4 to 6-ft. lengths. The cut lumber is not seasoned in the usual sense, and the practice is to measure it by weight and not by board feet as is generally the custom in the lumber industry.

The machines used to cut or shape lignum vitae are a combination of woodworking machines and standard metalworking ma-

chine tools. Standard wood-cutting machines are used to rip and cross-cut the logs, but conventional engine lathes, turret lathes, milling machines, and other such tools are employed to form the wood. The technique of machining this very hard dense wood is similar to metal-fabricating processes. High-speed-tool steels for the most part, are used in machining it, although some carbon-steel tools are used. A beginning is being made to use carbide-tipped tools for some applications.

The general machine-shop practice of roughing-out and then finish-machining parts is followed. Bores are normally drilled, bored, and reamed as in metalworking practice. In general, the material machines well. The lubricating gum is no doubt responsible for this. The wood machines better than maple or oak and does not burn or smoke at high machining speeds. Speeds and feeds are in the range of free-cutting brasses and aluminum.

Characteristics as a Bearing Material.

As a bearing material, its chief characteristic or asset is that it does not require lubrication of any kind. The Guaiac gum which forms one third of the material, is an excellent lubricating agent. This gum which gives the wood an oily or greasy feeling at room temperature and yet remains solid, will flow or become liquid when frictional temperature rises above 150 F. which in turn has a tendency to reduce the friction and therefore reduce the temperature. At normal temperatures this gum remains in the bearing indefinitely.

Under these conditions lignum vitae has a sliding coefficient of friction on steel of 0.16, as determined on an inclined slide without lubricant.

Most liquids when applied to the bearing act as a lubricant in the manner that oil or grease reacts to conventional bearing materials.

Since lignum vitae is a tree, it has growth rings or grain structure as have all other trees. This means that it has a laminated structure, and like most laminates its greatest strength is at right angles to the grain. To the trade this is called "end grain."

End-grain lignum vitae has been known to work successfully as a bearing at pressures of 2000 Psi. For this reason heavy-duty bearings are usually made of end-grain sections. Not all

lingum-vitae bearings are designed of end-grain materials or segmented sections. Most bearings made from lignum vitae are bushing or solid blocks as in general bearing-design practice.

In general, the proportions used for designing lignum-vitae bearings are those used for designing babbitt bearings. The bearings are designed for working pressures of 75 to 150 Psi. in the direction of the long grain and as high as 300 Psi. on the end grain. It should be noted that, because of the large clearances required in wood bearings, the working pressures are actually much higher than the same nominal-size babbitt bearing that is scraped or bored to close dimensions. Lignum-vitae bearings usually will not blue over 20 or 25 per cent of effective bearing area.

Lignum vitae works as well for sliding or reciprocating members as it does for revolving members. It makes good thrust bearings for heavy equipment. It is customary to make large thrust bearings in sections to obtain greater size.

Since lignum vitae does not contaminate, it is an excellent bearing material for use in food machinery and is used extensively for this purpose. It is not attacked by mild acids or brines and thrives very well under conditions where gritty particles are present in liquids. These grits become embedded in the grains of the wood and are then covered over by the gum.

Another characteristic of the material is that it has never been known to score, mark, or wear out the mating journal. Because of its physical makeup, it never seizes, as often happens in metallic bearing materials. In many bearing tests, where the wear of the hole is added to the wear of the shaft or journal, the resultant clearance is usually less than with other bearing materials.

Shortcomings in Physical Characteristics.

Because lignum vitae is a tree, it has all the deficiencies of a natural material, in that it is not uniform from tree to tree in strength, weight, resin content, or tendency to expand or contract. Like all woods, it will shrink and check. With lignum vitae, this tendency is exaggerated, for not only will the material shrink or expand excessively but it does not shrink uniformly as to direction or amount, and never stops its movement with age, but will always expand and contract noticeably with changes of humidity or temperature.

As an example, a bearing with a ⅛-in. hole and a ⅛-in wall section will shrink as much as 0.015 in. in a 10-day period following the machining operations. A 3-in.-diam. bearing with a wall thickness of ½ in. will shrink the same amount in the same period. It is easy to understand that the shrinkage will be greater in one direction than in another, because of the laminated structure of the material. Because of the unstable conditions, lignum vitae is only used in bearings or installation where relatively large clearance and large tolerances are permitted. This in turn relegates it to slow-speed applications. In spite of its confinement to the nonprecision-bearing class, lignum vitae still has applications where its ability to perform under adverse conditions without attention or maintenance has made its replacement by its more physically stable competitors almost impossible.

III—Properties and Applications of Laminated Phenolic Bearings.

By L. E. Caldwell and John Boyd.

The use of laminated phenolic bearing materials in a wide range of installations over a period of years has shown them to possess important properties which are particularly useful in many applications. The advantages and limitations of these materials, which appear in proper perspective only after prolonged use, have now been fairly definitely determined. The purpose of this paper is to outline and discuss the properties, and certain of the more important applications of these materials.

Manufacturing Considerations.

Laminated phenolics fall in the general classification of thermosetting plastics. This group of materials includes those which cure, or harden upon the application of heat to form a horny substance which, for practical purposes, will not soften or fuse on further heating.

Phenolic laminations, in general, are made by treating filler sheets of either paper or fabric with a solution of phenolic resin and then carefully heating to remove the solvent and partially set the resin. The resin in this intermediate stage is neither infusible nor wholly soluble. These laminations are then super-

imposed on one another and subjected to further heating, called curing, to bond them together into a more or less homogeneous mass and to complete the setting or hardening of the resin. In some cases, it is desirable to add other filling materials. These are usually in powdered form and are generally incorporated into the laminated structure by depositing them on the bare filler sheets before treating with resin, or by suspending them in the resin solution before treating. By changing the details of processing and by selecting proper fillers, the properties of the finished material may be varied to meet the requirements of different applications. The methods of handling laminations in the last stages of manufacture are dictated by the type of bearing being made (see Fig. 1).

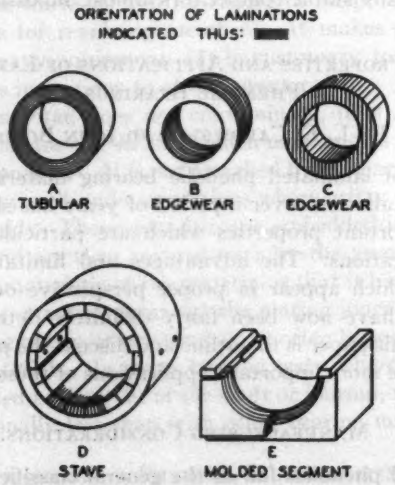


FIG. 1 VARIOUS TYPES OF LAMINATED PHENOLIC BEARINGS

Laminated phenolic tubing is formed by winding the treated material under tension onto a heated steel mandrel. As each successive layer is wound on, it is bonded by fusing the uncured resin as it passes over heated pressure rolls. The final curing is then carried out on the steel mandrel by one of the following methods: (1) heating in an oven without any additional external pressure, or (2) molding in a cylindrical mold. When the mandrel

is stripped from the cured mass, the rough tubing is ready for finishing operations. The rolled tubing, as produced by the first method of curing, is less dense and has somewhat lower physical properties than the molded tubing produced by the second method, since the only pressure involved in its manufacture is that produced by winding under tension and the shrinkage which occurs during curing. The chief limitation of molded tubing is the size and quantity of the equipment required to produce it, and hence the major portion of laminated phenolic tubing is produced by rolling only.

Laminated phenolic plate is produced by curing stacks of the treated laminations between the heated platens of a hydraulic press. In this case, pressures of from 1000 to 2000 Psi. are applied to the material to produce a mass of maximum density and mechanical strength for a given type of material

Molded laminated shapes, such as molded radial bearing segments, etc., are in essence curved laminated plate for which the molding surfaces are machined to the correct diameter or shape instead of being flat platens as in the case of flat plate.

The material just referred to as "cured" is at the end point of a chemical reaction accelerated by the application of heat in so far as practical considerations are concerned. Further reaction or cross-linking of the resin molecules can be produced by additional heating, but this reaction proceeds at a very slow rate and in general, does not further improve the properties of the material. The cross-linking of molecules can and will continue, at a much reduced rate at room temperature, giving rise to a slight after-shrinkage of the material. Theoretically, the reaction could continue to the point where all the resin in a given piece became one molecule.

Physical Properties.

A material manufactured in the foregoing manner exhibits directional properties. In other words, the physical strength of the material depends upon the direction of orientation of the laminations with respect to the applied forces. The greatest tensile strength is obtained when the material is stressed parallel to the laminations, and the greatest compressive, flexural, and impact strengths are produced when the material is loaded perpendicular to the laminations. A somewhat greater Rockwell M

TABLE 1 PHYSICAL PROPERTIES OF LAMINATED PHENOLIC BEARING MATERIALS

Property	Direction[a]	Plate or molded laminated shapes					Rolled tubing		
Fabric weight, oz per sq yd	...	3	6½	8	15	50	3	6½	6½
Resin, per cent	...	60	52½[b]	60	60[b]	40	60	55	60[b]
Specific gravity	...	1.35	1.38	1.38	1.72	1.46	1.30	1.30	1.28
Moisture absorption at saturation, per cent increase in weight	...	3	7	4	6	13			
Tensile strength, psi	WG	7000	11500	11500	9000	7000			
	CG	6000	7000	7000	10000	6000	8500	8000	6500
Elastic modulus, tension, 1000 psi	WG	800	1000	1000	900	600			
	CG	800	600	600	900	600			
Compressive strength, psi	F	40000	36000	37000	38000	32000			
	E	33000	25000	25000	20000	17000	23000	20000	19000
Elastic modulus, compression, 1000 psi	F	500	500	500	700	400			
	E	500	500	400	500	200			
Flexural strength, psi	FWG	17500	18500	19000	15200	12200			
	EWG	15500	17500	18000	14800	11700			
Shear strength, psi	FWG	11000	7000	9500	9200	8400			
	EWG	13000	7000	10500	10100	8500			
Izod impact strength, ft-lb per in. width	FWG	0.8		4	4.7	7.4			
Bond strength, lb[c]	EWG	1600	1200	1600	1900	1700			
Hardness, Rockwell M[d]	F	115	106	112	102	90			
	E	108	98	109	97	74			

[a] G grain (parallel to warp of fabric). CG cross grain. F flatwise (perpendicular to laminations).
WG with grain. E edgewise (parallel to laminations). EWG edgewise with grain.
FWG flatwise with grain.

[b] Including mineral filler.

[c] Pounds to split 1 in. × 1 in. × ½ high specimen with 1-cm ball.

[d] 100 kg load, ¼ in. ball.

hardness is found when measured perpendicular to the laminations than when measured parallel to them. Table 1 lists a number of the physical properties of some of the more common types of laminated bearing materials.

The weights of fabric used affect the physical properties of the various grades of material as may be seen in the table. This is due to several factors, the primary one being the lesser degree of impregnation achieved with the heavier fabrics. Other factors which are less important are the construction of the fabric, such as the tightness of the weave, the twist of the yarn, and the like, and whether the fabric is bleached or unbleached.

Laminated phenolic bearings, depending upon application, employ fabrics ranging in weight from 3 to 50 oz. per sq. yd. of unbleached cloth or belting. The static physical properties of these materials, shown in Table 1, do not have nearly as wide a range as do their dynamic physical properties, as evidenced by their Izod impact strengths.

As may be concluded from the foregoing, the better impregnation and the higher resin percentage obtainable with fine-weave fillers, as compared to that of coarse-weave fillers, determine to a great extent the amount of variation that exists between the various grades. As the percentage of resin used in a laminate increases, the modulus of elasticity rises. This factor is of importance in applications requiring a minimum of deflection.

Other Factors Affecting Use as a Bearing Material.

In addition to the physical or structural properties of a prospective bearing material, many other factors must be considered to determine whether a bearing made to a given design from such a material will perform satisfactorily in a given application. In doing this, the bearing, the lubricant, and the journal, rather than just the bearing alone, must be taken into account.

Speaking generally and including other bearing materials as well, it is usually not possible to separate all the factors which contribute to the success or failure of a bearing-lubricant-journal combination into independent variables which can be evaluated specifically. However, certain more or less fundamental requirements may be set down.

In general, such a combination should possess the following: (1) compatibility, (2) stability, (3) adaptability, (4) economy,

and (5) any additional special properties required for the application at hand.

These factors determine whether the components will operate together under varying conditions, whether they will resist physical, dimensional, and chemical deterioration, whether they possess the ability to adapt themselves to the usual hazards of operation, and whether their use permits a saving of space, time, or money. These requirements are listed and enlarged upon in Table 2.

Since requirements vary from one application to another and since no bearing combination can excel in all the properties listed in the table, the choice for a given application must be based upon satisfying as many conditions as possible.

The manner in which laminated phenolic bearings, together with various lubricants and journals, measure up to these criteria may be stated as follows:

Compatibility. Laminated phenolic materials inherently possess the property of being able to operate well with steel and bronze journals and with oil, water, or certain other substances as lubricants. Bearings and journals made from such materials show little tendency to weld or adhere to one another. Consequently they have high resistance to seizures.

TABLE 2. OTHER REQUIREMENTS FOR A BEARING-LUBRICANT-JOURNAL COMBINATION.

Compatibility: Ability of components to operate together under varying conditions		Resistance to seizure and galling at all loads, speeds and temperatures encountered in service. Ability to dissipate frictional heat
Stability: Ability of components to resist deterioration	Physical	Resistance to change in physical properties due to heat, load, load cycle, time, etc. Resistance to fatigue, shock, hammering-out, etc.
	Dimensional	Resistance to wear, swelling, shrinking, warping, etc.
	Chemical	Resistance to corrosion, oxidation, or other reactions between components or adjacent material

Adaptability: Ability of components to adapt themselves to usual hazards of operation	Ability to adjust for slight misalignment, small inaccuracy, etc. Conformability
	Ability to absorb or eliminate foreign materials. Imbeddability
	Ability to operate for short periods without lubricant, etc.
Economy:	Economy of space, power loss, initial cost, maintenance, repairs, etc.
Special Properties:	Quiet operation, lightweight, cleanliness of operation, etc.

The hardness of phenolics is ordinarily expressed in terms of the Rockwell M scale and usually runs around 100 (see Table 1). This is equivalent to approximately 40 Brinell, as measured with a 500-kg. load. Since the hardness of annealed mild steel is of the order of 115 Brinell (500-kg. load), it may be seen that shaft scoring is very unlikely to occur. Usually the journals become well polished in service.

The presence of fillers made from finely powdered hardening agents, used in certain applications such as roll-neck bearings, produces a high finish on the journals.

Although phenolics have a very low thermal conductivity, the important consideration in so far as the bearing combination is concerned is the ability to dissipate frictional heat. Thus by water-flooding, as in roll-neck bearings, large amounts of heat may be removed even though the bearing is made of a rather efficient thermal insulator.

Polishing the journal before it is put into operation helps to reduce the bearing temperatures because it lessens the amount of frictional heat developed. In some water-lubricated bearings, where unusually severe conditions are encountered, it is desirable to use other lubricants along with the water. Many reversing mills, particularly blooming and slabbing mills, show better results when small amounts of grease are injected at regular intervals.

Stability.

(*a*) *Physical.* Because of their natural resilience, phenolic materials are highly resistant to deterioration resulting from

fatigue and shock. They make excellent rudder pintle bushings because they are not hammered out like many other materials under the heavy shock loads.

(*b*) *Dimensional.* The immersion of laminated phenolics in either water or oil does not produce any appreciable change in their mechanical strength. However, since they contain organic fibers which dilate in the presence of certain liquids, slight changes in dimensions may result from contact with the lubricating medium. In the case of lubricating oils, the changes are so slight as to be negligible, but with water or aqueous solution, they are measurable and in some applications must be considered if an adequate design is to be produced. In this connection, it is important to note that the term "per cent miosture absorption," frequently used in describing the properties of laminated materials, does not give an accurate conception of dimensional stability, since it is based on change in weight rather than change in volume.

In general, the finer-weave fabrics produce materials with better dimensional stability because of the smaller amount of moisture which they absorb. Fig. 2 shows the dimensional changes in a marine-grade phenolic material. It will be noted that water produces an expansion of from 2 to 3 per cent perpendicular to the laminations and from 0 to 0.3 per cent parallel to the laminations. The curve also shows the relatively small expansion obtained with oil. The negative slope of the right-hand portion of the curves for 200 F is due to a small superimposed shrinkage caused by maintaining the material at the elevated temperature.

The curves in Fig. 2 are based upon free expansion in all directions. If the material were restrained perpendicular to the laminations, as in the case in some bearings, the expansion parallel to the laminations would be much greater, sometimes as high as 3 per cent.

Phenolics have high wear-resistance qualities and have been found to outlast considerably other materials in many applications.

(*c*) *Chemical.* One of the important characteristics of phenolics is their resistance to chemical attack. They are unaffected by water and oil and are practically uninfluenced by dilute acid and caustic solutions.

Adaptability. Laminated phenolics have a high degree of adaptability. With an elastic modulus of 1×10^6 Psi., they are better able to conform to journal inaccuracies and misalignments than babbitt and bronze, which have moduli of 5 and 9 times as large, respectively.

They also possess a high degree of embeddability. This property is employed to advantage on ship stern-tube propeller-shaft bearings, which must operate in water containing sand or sediment.

Economy. Phenolic bearings frequently show considerable power saving because of the relatively low friction obtained. Fig. 3 shows the coefficient of starting friction as a function of load for three types of stern-tube bearings. Fig. 4 shows the coefficient of friction versus rubbing speed for a stern-tube bearing with 40 Psi. load, and a roll-neck bearing with 4000 Psi. load. Fig. 5 shows the machine upon which the data for Figs. 3 and 4 were obtained. The design is based upon equipment developed at the Naval Experiment Station.

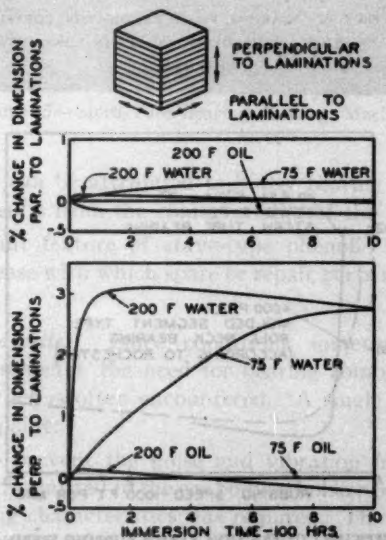


FIG. 2 DIMENSIONAL CHANGES IN MARINE-GRADE LAMINATED PHENOLIC MATERIAL DUE TO WATER AND TO OIL IMMERSION

Actual installations may be expected to show some variation from the foregoing values, depending upon the condition of the journal and bearing surfaces, but in many installations the low coefficient of friction obtained by water lubrication has resulted in considerable power saving compared to metal bearings. Part

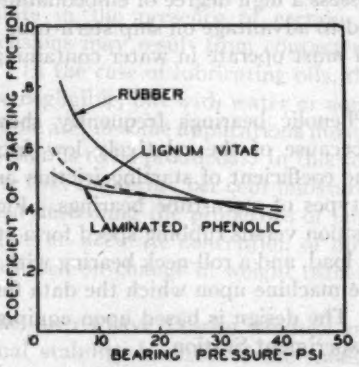


FIG. 3 EFFECT OF BEARING PRESSURE ON THE COEFFICIENT OF STARTING FRICTION, 6-IN-DIAM × 12-IN-LONG STERN-TUBE BEARINGS

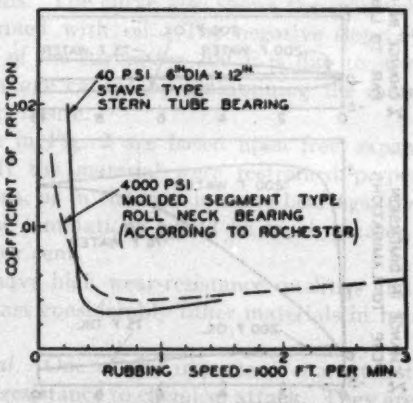


FIG. 4 COEFFICIENT OF FRICTION VERSUS RUBBING SPEED; WATER-LUBRICATED LAMINATED PHENOLIC STERN-TUBE AND ROLL-NECK BEARINGS

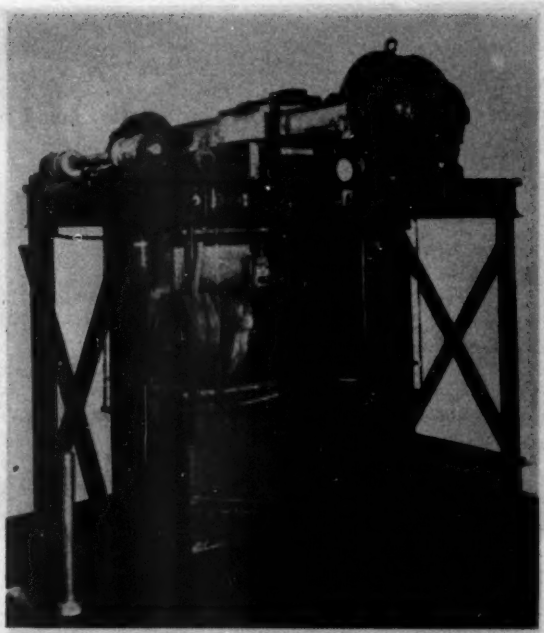


FIGURE 5.—Stern-Tube-Bearing Testing Machine.

of this saving can be attributed to better bearing practice, but much of it results from the characteristics of the material.

An important feature of stave-type phenolic bearings (see Fig. 1) is the ease with which spare or repair parts may be stored and fitted.

Special Properties. Since conditions sometimes establish unusual requirements, the need for bearing combinations with special properties is often encountered. A single example will illustrate the point.

In order to prevent the noise and vibration from streetcar trucks being transmitted to the car body, a bolster bearing having noise-deadening characteristics was required. The problem was satisfactorily solved by the use of laminated phenolic bolster cup bearings (see Fig. 6).

FIGURE 6.—Laminated Bolster Cup Bearing for P.C.C. Car; 12 in. OD.

TYPES OF LAMINATED PHENOLIC BEARINGS.

The types of construction used for laminated phenolic bearings is determined primarily by the kind of service to which they will be subjected. Secondary considerations are, of course, manufacturing facilities, interpreted in the light of the number of identical pieces required. The main types are shown in Fig. 1.

Tubular bearings (type A) are machined from laminated tubing stock. They are used principally under conditions requiring a complete bushing. They are available in any size up to 48 in. ID (see Fig. 7).

Edgewear bearings (types B and C) are machined from laminated plate stock. They are intended for light-duty service. Type B bearings are normally 2 in. or less in length and are generally used as guide bearings.

FIGURE 7.—Laminated Rudder-Stock Bearings for Cargo Vessels; 18 in. ID.

FIGURE 8.—Laminated Stern-Tube Bearing Installed in a Cargo Vessel; 15 in. ID.

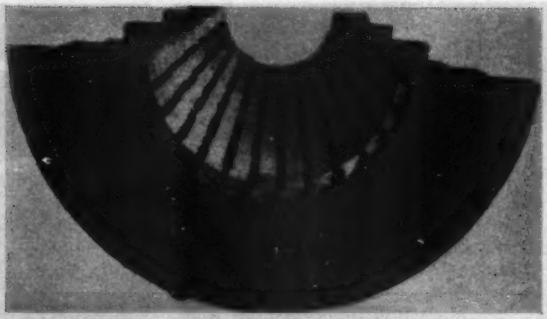


FIGURE 9.—Laminated Guide Bearing for Water-Wheel Turbine; 24 in. ID. (one half of bearing shown)

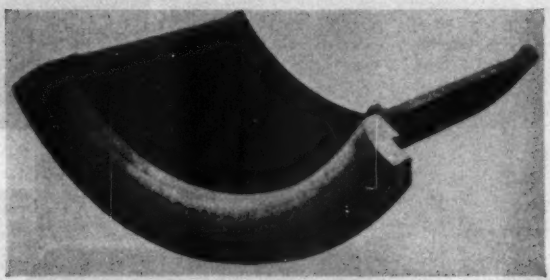


FIGURE 10.—Laminated Roll-Neck Bearing for Three-High Breakdown Mill; 22 in. ID.

Type C bearings are also machined from plate stock. They are available up to 3 in. ID in any length that is practical to bore and are used primarily for small-craft stern-tube bearings.

Stave bearings (type D) are fabricated from laminated plate stock and can be made in any size from 3 in. ID up. They are used mainly for stern-tube and rudder-stock bearings on ships and for guide bearings on vertical water-wheel turbines (see Figs. 8 and 9).

Molded segment bearings (type E) are molded to size. They are normally used in conjunction with a thrust plate which can be either molded or fabricated from plate stock. They find their greatest use as roll-neck bearings in the steel industry (Fig. 10).

Bearing Clearance.

In establishing the clearance which should be provided when machining a laminated phenolic bearing, it is necessary to consider the actual running clearance desired and the additional clearance necessary to compensate for any swelling that may take place after machining.

In the case of partial or open-face bearings (bearings with less than 180-deg. arc, such as roll-neck bearings), it is usually unnecessary to make any allowance for swelling in determining the clearance.

In the case of full-or bushing-type bearings (bearings which completely surround the journal), however, it is essential that swelling be taken into account. A consideration of the design of a marine stave-type stern-tube bearing will illustrate the manner in which this is done.

Since such bearings usually have low bearing pressures (of the order of 25 Psi.), the staves are installed so that the load is carried on the edge of the laminations rather than flatwise. This construction gives more uniform wear properties.

In installing the staves, it is desirable to machine the individual pieces and to bore the assembled bearing dry, thereby eliminating the difficulties attendant upon saturating the phenolic material and keeping it saturated until ready for use.

If the staves in a 6-in. stern-tube bearing with ½-in.-thick staves, were to swell freely, the circumference would increase about 0.6 in., and the thickness of the staves about 0.0015 in. (3 per cent tangentially, perpendicular to the laminations and 0.3 per cent radially parallel to the laminations).

In order to keep the radial swelling as small as possible and to make removal of the shell from the housing easier by minimizing the shell distortion due to swelling forces, stern-tube bearings are usually fitted with elastic or spring staves as shown in Fig. 11. These staves absorb the circumferential swelling with a minimum of resulting force and permit the installation of such bearings with small clearance allowance for swelling. Figs. 9 and 12 show a slotted shell in which each slot is fitted with its own spring stave.

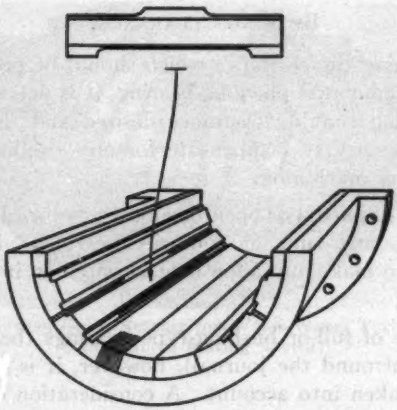


FIG. 11 SOLID-PACKED STERN-TUBE BEARING SHOWING SPRING STAVE

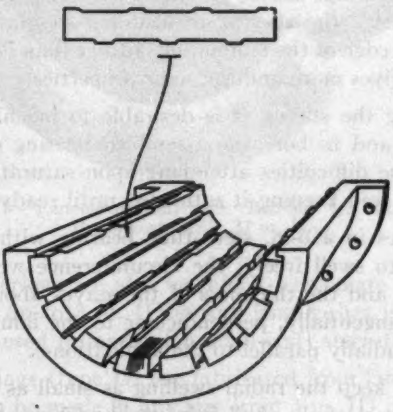


FIG. 12 SLOTTED-SHELL STERN-TUBE BEARING FITTED WITH SPRING STAVES

ROLL-NECK BEARINGS.

The first development work on laminated phenolic roll-neck bearing material was carried out on a medium-weave (12-oz. per sq. yd.) fabric. In the early trials, the material was more or less naturally turned on edge in the same manner as the lignum vitae

it was being used to replace. The first application was on a rod mill. It was found that the bearing worked satisfactorily with water lubrication, and additional installations were made with the same results until the bearing pressures became too great for this on-edge type of construction. Trials were then made with the same material turned flatwise in order to take advantage of the greater compressive strength of the material only to find that with the same loading and speed conditions the material would blister as the result of local overheating.

This objection to flatwise-constructed laminated phenolic bearings was first overcome by the use of heavy fabrics. The fabrics (30 to 50 oz.) used were woven in such a manner as to present ends of some threads in the bearing surface. This, in effect, gave a partial edge-grain surface with the strength of flatwise construction. The explanation of the difference between the edge-grain and flatwise operating characteristics is that the former more readily allows the escape of gases produced under the surface by local overheating than does the latter. Needless to say, such overheating is a self-aggravating condition once it has started and, if the temperature reaches the point at which gases or steam are produced and trapped beneath the surface, the degree of self-aggravation increases to the point of severe heating and failure.

The heavy-weave materials will carry high loads even at slow speeds and will also withstand severe impact loading without the hammering-out experienced with babbitt. Under such service as is experienced in blooming, slabbing, and some roughing-mill service, they still give the best performance of any of the laminated bearing materials available. However, as a result of their comparatively low resin percentage, their rate of wear is relatively great after the initial hard surface is worn off.

The next step in the development was the addition of graphite as a mineral filler to the materials already available. This lowered the degree of bonding between the laminations and consequently produced a somewhat softer material with correspondingly lowered strength. These materials have proved satisfactory for some applications.

After the graphite materials were tried, other mineral fillers which would increase the abrasion resistance of the laminated materials were introduced. It was found that medium-weave

TABLE 3 TYPICAL LAMINATED PHENOLIC BEARING APPLICATIONS

Bearing application	Type[a]	Size range, in.	Fabric weight, oz per yd	Resin, per cent	Lubricant	Diametrical clearance, in.[b]	Principal reasons for using laminated phenolic bearing material
Roll-neck bearing	E	3–30	15–50	40–60	Water or emulsion	0–1/2	Longer life—power saving due to lower friction—lower-cost water lubrication—greater cleanliness of operation—better holding of gage due to less wear
Ship stern-tube bearing	D	3–26	8	60	Water	0.001 per in. diam. over 5 in. diam.	Longer life—greater ease of handling and installing—higher load capacity, particularly with impact loads—lower friction—greater corrosion and decay resistance—lower journal wear—greater local availability
Ship rudder-stock. Pintle bearing	A, D	3–36	3–8	55–60	Grease, water	0.001 per in. diam. over 5 in. diam.	
Small-craft stern-tube bearing	C	1/2–3	8	60	Water	0.005	
Centrifugal pump bearing	A, B, C	1/2–4	3–8	60	Liquid pumped	0.005	Longer life—better lubrication with liquid (water, gasoline, chemical solutions, etc.) pumped
Water-wheel turbine guide bearing	D	4–24	8	60	Water	0.005	Longer life—lower friction—no decay—less journal wear
Ball-mill bearing	A, E	15–48	6 1/2–15	55–60	Water, emulsion, water and grease	0.015–0.030	Longer life—higher load capacity—lower friction—lower lubricant cost
Aircraft landing-gear bearing	A	2–15	3	60	Oil	0.001 per in. diam. over 5 in. diam.	Lighter weight—satisfactory dimensional stability and load capacity
Railway bolster-cup bearing	Molded cone	...	6 1/2	53	Grease	...	Longer life—lower noise and vibration transmission

[a] See Fig. 1. [b] Running clearance; does not include allowance for swelling.

(12-oz.) fabric laminates, including finely powdered hardening agents, when applied flatwise would give much better service in roll-neck bearings operating at 200 or more ft. per min. than did any of the previous laminates tried. Below 200 Fpm. slip velocity these materials show a tendency to over heat and seize, but at speeds great enough to bring sufficient cooling water to the bearing surface, they polish or lap the journal surface to a high finish and thereby lower the operating coefficient of friction below that of straight laminates. Such bearings require a larger quantity of cooling water than do the other grades of materials. Where operating conditions permit their use and a slight amount of journal wear is not objectionable, they give the best service of any laminated phenolic material available.

Table 3 is useful in illustrating how the advantages of laminated phenolic materials are applied and their limitations taken into account in a number of typical bearing applications.

Conclusions.

The foregoing discussion has attempted to describe the properties of laminated phenolics as bearing materials. Their limitations will, of course, exclude many applications, but their numerous advantages will undoubtedly widen their present fields of application.

IV—Reduction of Wear in Rubber Stern Bearings by Flow-Controlled Continuous-Film Method.

By R. D. Smeallie.

During the recent war, landing-craft propeller-shaft bearings and shafting were a costly and time-consuming maintenance problem. In peacetime, the condition is paralleled in many parts of the world where vessels are forced to operate in shallow waters heavily laden with sand, silt, coal screenings, and coral, all of which are conducive to excessive shaft or shaft-liner and bearing wear.

In attacking the problem of minimizing maintenance from this natural hazard, it became apparent at once, that, not only was a complete mechanical change inevitable, but the technique of lubrication also must be overhauled to increase greatly the sand-ejection rate within the bearing.

The first mechanical change toward the objective was an important step in shaft conservation and consisted of combining the shaft sleeve and bearing as an integral one-piece unit. In pursuing this step, the shaft sleeve with its molded rubber covering was locked to and rotated with the shaft, the "rotary journal" using for its opposite frictional surface a stationary renewable bronze bushing. This type of rubber bearing, although not of a continuous-film construction at that time was used successfully throughout the war for propeller-shaft bearings on landing craft.

The next step was based on a study of hydrodynamics which brought to light a little-used law of physics involving action of a submerged shaft rotating under water at various loads and speeds.

As a result of the study, it was determined that the internal action of a submerged bearing took on a definite pattern of a vacuum-pressure cycle which when analyzed was observed to set up a dual action of pressure and vacuum for each 180 deg. of a revolution. This double cycle has now been developed to produce several characteristics which heretofore seem to have been entirely neglected in the design of marine stern bearings. It became clear that the primary requisite was to block all film "pressure escape points" having their origin within the loaded section of the bearing. The first apparent advantage was the formation of a "continuous" lubrication film which, in effect, allowed the build-up of a perfect water wedge. This was highly desirable inasmuch as it afforded a very high degree of "shaft flotation." Separating the frictional surfaces by this uninterrupted water wedge reduced materially the coefficient of friction at practically all speeds with particular emphasis at low speed where shaft and bearing damage is at its most critical point.

A definite proof of complete shaft flotation was evidenced by the fact that, instead of the maximum-wear point being 12 to 15 deg. ahead of dead center (in direction of rotation), the effected water wedge had actually picked up the shaft and changed its maximum point of wear to approximately 15 deg. after passing through dead center. Thus a change of nearly 30 deg. took place in the maximum-wear point between the continuous-film and noncontinuous-film method of lubrication.

Following further reactive results, it was found that as soon as the shaft speed was sufficiently high to effect shaft flotation,

an important and pronounced sequence occurred. For example, assuming a point diametrically opposite the maximum load, the generated film pressure between frictional surfaces was zero. Progressing in the direction of rotation, consecutive readings taken on a compound gage showed increasing pressure until it arrived at its greatest value directly in the maximum loaded area. This action is a long-recognized lubrication fundamental which has been disregarded, almost without exception, in the present-day design of stern bearings.

Passing through the next 180-deg. cycle, the pressure, within a few degrees after having passed through the maximum loaded area, had reverted to a maximum vacuum which decreased in the direction of rotation until it subsided completely at the original starting point. This dual cycle was repeated on each revolution of the shaft and merely reversed the pattern of action when the shaft was rotated in the opposite direction.

The following step, however, was an entirely new conception of rapid sand removal and a direct result of applying theories known to exist in the double pressure-vacuum cycle briefly described.

Putting the Pressure-Vacuum Cycle to Work.

By placing two axial-flow control grooves running from end to end on the inside periphery of the mating bronze bushing,

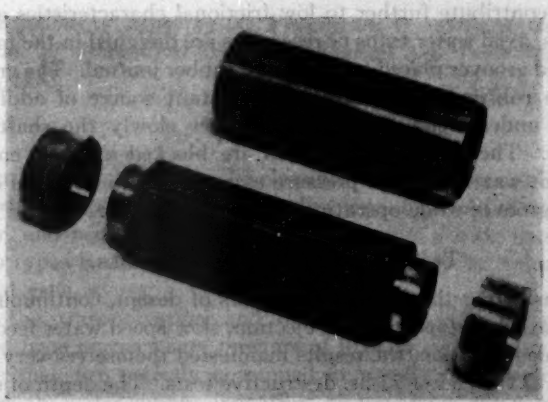


Figure 1.—Rotary Rubber Bearing.

one in the pressure area and one in the vacuum area, advantage was taken of the hydrodynamically developed pressure and vacuum values. Sand and water were taken in on the vacuum cycle while sand and water were also discharged under pressure during the pressure cycle. Both actions are continuous, once the shaft is rotating, and will duplicate the action in either direction of rotation.

Part of the phenomenon occurring in the action somewhat defies a clear mathematical explanation inasmuch as the water flow in the pressure-control groove is a right-angle component of the circumferential water flow. Existing proof of the action may be cited when water levels are dropped to the center line of the bearing at which point water can be observed being ejected from the pressure groove with considerable force while the vacuum grooves of course show no discharge. When testing large bearings, the vacuum action becomes sufficiently violent to draw a person's hand against the intake area at the ends of the bearing.

One of the outstanding characteristics of the continuous-film method is its reduced low speed coefficient of friction. For example, at 100 Rpm. with a load of 30 Psi., the frictional torque is 50 per cent less than for the interrupted-film method. Frictional horsepower, by the same token, had a ratio of 2 to 1 in favor of the continuous-film method.

To contribute further to low frictional characteristics at low speeds, axial water traps or reservoirs are designed in the form of molded grooves placed in the rotary rubber journal. The grooves in the rubber tend to supply a constant source of additional water under pressure regardless of how slowly the shaft may rotate. The slow-speed grooves are blocked at both ends to prevent water and film pressure escape through the ends of the rotor grooves while operating in the loaded area.

Tests Demonstrate Advantages.

Considering the four major factors of design, continuous-film lubrication, automatic sand ejection, slow-speed water feed, and shaft conservation, the results manifested themselves very effectively at the end of 72-hr. destructive tests. The depth of scouring in the metal bushing took on an even depth which was

approximately 70 per cent shallower than for the noncontinuous-film type, score lines being barely perceptible to the fingernail. When running parallel tests on several types of interrupted-film nonflow control-type bearings, the wear progressed through to the housing, causing the test to be discontinued as inoperative. The continuous-film flow-controlled bearing still had 42 per cent life remaining, practically all wear being taken on the renewable metal bushing and less than 0.001 in. on the rubber. Because of its integral rubber-covered shaft sleeve, the shaft was preserved in its original state.

Continuous-film lubrication is of unquestionable engineering value and has been used for many years in the operation of high-speed turbine bearings, reduction-gear units, and for various heavy-duty industrial bearing uses. This method of lubrication, combined with automatic sand ejection, slow-speed water feed to frictional surfaces, and complete preservation of shafting, bids fair to the establishment of a new low in maintenance costs.

CENTRIFUGAL AND OTHER NON-POSITIVE PUMPS.

The following paper by R. Pennington is reprinted from the Institute of Mechanical Engineers Journal and Proceedings for June, 1948.

Application Groups.

Centrifugal pumps may be classified in five main application groups: transportation and distribution, disposal, circulation, power, and marine. In each of these groups a wide variety of pump types is used and one or more types may be common to several groups.

In the transportation and distribution group, liquids are moved from places where they are of little or no value to places where they are usefully consumed or employed. Typical examples are water supply, irrigation, oil pipe-lines, fire service.

In the disposal group, liquids are moved from places where they are harmful or inconvenient, to suitable points of disposal; for example, mine drainage and sinking, land drainage, sewage and storm water, and dredging.

Pumps in the circulation group maintain a continuous flow of liquid through a system, a function as vital to the particular application as the heart is to man's life-carrying stream. In this category are pumps for boiler feeding, condensate extraction, condenser circulation, boiler circulation and boiler recirculation in steam power stations; district heating, oil refineries, chemical process plant, sugar refineries, food processing, breweries, paper mills, central heating, heat pumps, etc.

In the power group, pumps generate pressure to do work directly as hydraulic pressure operating a machine, by conversion of pressure to velocity, or to provide energy in some other manner. Typical examples are hydraulic power for presses, shears, and furnace control in steelworks; lifts, cranes, and docks; baling and pressing of cotton and jute; descaling (steel mills, boiler tubes, etc.); hydraulic sluicing (china clay mines, gold mines, etc.); oil fire-fighting; hydraulic couplings and torque convertors; water-power storage pumps; and rocket power units.

The marine group includes all the many and varied types used on board ship for such services as sea-water and fresh-water circulating and cooling, ballast, sanitary, bilge, oil fuel, lubrication, and general service.

Present Standard of Attainment.

Centrifugal pumps have attained a high standard of general performance and reliability. In this address it is possible to present only a brief outline of a few of the features of design and construction which may be of special interest, as follows:—

Cavitation. Pump designers have known that cavitation could be avoided by co-ordinating the design with that of a pump of similar type and of known characteristics operating under conditions of dynamic similarity. At least twenty-one factors are associated with cavitation and by a process of analytical research these factors have been correlated so that the performance of a pump can be predicted without reference to a model. The results are summarized here, but the subject will be

The controlling factors fall into three general groups:—

(*1*) *Hydraulic Factors.*

Rate of pumping or pump output	Q
Total generated head (single stage and multi-stage)	H
Speed	N
Pump efficiency	e
Number of stages	n
Net head in first stage	h
Depression head	h_{dp}
Inlet size of pump	$L_i = Q^{\frac{1}{2}}/h_{dp}^{\frac{1}{4}}$
Head at pump inlet branch	h_i
Specific speed	N

(*2*) *Physical Properties of Liquid, etc.*

Temperature of liquid	$t°$
Vapour pressure of liquid	h_v
Kinematic viscosity of liquid	ν
Specific gravity of liquid	S_g
Barometric pressure	h_b

(*3*) *Design Features of Pump.*

Single-inlet or double-inlet impeller.
Design of pump inlet passage to impeller.
Design of impeller vanes at inlet.

Diameter of pump shaft	d_s
Diameter of impeller hub	d_h
Diameter of impeller eye	d_e

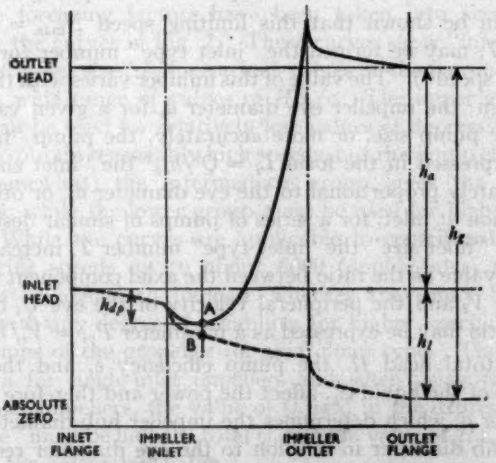


Fig. 1. Hydraulic Gradient through a Centrifugal Pump; Depression Head

discussed more fully in a paper which I hope will be published in the near future.

In general it is desirable to run a centrifugal pump at the highest speed consistent with good efficiency and freedom from cavitation in order to ensure minimum dimensions and lowest cost. Let us see how this maximum permissible speed may be determined.

Fig. 1 shows diagrammatically a typical hydraulic gradient through a centrifugal pump operating on normal duty. The dotted curve shows the fall in head between the inlet and outlet flanges assuming it were possible to ignore the head generated by the impeller, and the vertical intercepts between the dotted and full curves indicate the head generated in passing through the impeller. Thus the full curve represents the actual head at any point in the pump, and it will be noted that the trough of the depression occurs at a point just beyond the inlet edge of the impeller blades as shown by point A. The fall in head between the inlet flange and point A may be called appropriately the "depression head" h_{dp}. If at this point the absolute head is equal to the vapor pressure of the liquid at the working temperature, the pump is operating on the cavitation limit and at the maximum permissible speed for the particular output Q.

It can be shown that this limiting speed $N_{max} = T_i h_{dp}^{\frac{3}{4}}/Q^{\frac{1}{2}}$ where T_i may be termed the "inlet type" number (or "suction specific speed"). The value of this number varies with the quality of design; the impeller eye diameter d_e for a given value of Q; and the pump size, or more accurately, the pump "inlet size" L_i. Expressed in the form $L_i = Q^{\frac{1}{2}}/h_{dp}^{\frac{3}{4}}$ the "inlet size" is approximately proportional to the eye diameter d_e, or other linear dimension at inlet, for a series of pumps of similar design. For a given "inlet size" the "inlet type" number T_i increases up to a peak value as the ratio between the axial component of flow in the eye V_i and the peripheral velocity of the eye U_i decreases. This ratio may be expressed as a parameter $P_{pi} = V_i/U_i$.

The total head H, the pump efficiency e, and the specific gravity of the liquid S_g, affect the power and therefore the shaft diameter d_s which determines the impeller hub diameter d_h. A large hub diameter in relation to the eye diameter reduces the "inlet type" number.

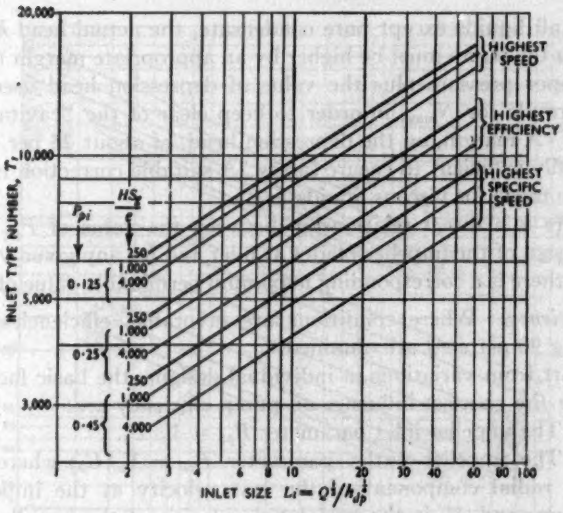


Fig. 2. Relation between Inlet Type Number and Inlet Size
Values are for single-inlet impellers. For double-inlet impellers total Q must be halved and HS_g/e doubled.

All the foregoing factors have been taken into account in preparing the curves in Fig. 2. These curves show "inlet type" T_i against "inlet size" L_i for various values of the parameter P_{pi} and for a wide range of values of HS_g/e. The upper group of curves may be used for relatively low values of depression head in order to obtain reasonably high speeds, but the pump efficiency suffers somewhat; the intermediate group gives the highest efficiences; while the lower group must be used for high specific speeds. While the curves are applicable to centrifugal pumps with specific speeds up to about 6000 (double-inlet) there is insufficient evidence to indicate whether they may be used directly with any degree of reliability for higher specific speeds or for pumps of the propeller (or axial-flow) type. The curves are drawn for single-inlet impellers and when using them for double-inlet impellers, the value of Q used in the expression for "inlet size" must be half the total Q, but the value of HS_g/e must be twice the calculated value.

For all liquids except pure condensate, the actual head h_s at inlet to the pump must be higher by an appropriate margin than the vapor pressure plus the value of depression head used in the formula for N_{max}, in order to keep clear of the "cavitation area". A margin, on the depression head, of about 25 per cent is usually sufficient to ensure safety. A suitable correction must also be made for viscous liquids.

There is a direct relationship between the value of P_{pi} and the design of the impeller blades at inlet and for any given value of P_{pi} there is a corresponding maximum permissible value of N_s.

Efficiency. Where conditions are favorable, efficiencies exceeding 90 per cent are obtained.

Apart from variations in individual designs, the basic factors having the greatest influence on pump efficiency are:—

(1) The impeller inlet parameter $P_{pi} = V_i/U_i$.

(2) The impeller outlet parameter $P_{po} = V_o/U_o$; where V_o is the radial component of the exit velocity at the impeller periphery and U_o is the peripheral velocity of the impeller at outlet.

(3) The pump size, or more exactly, the pump "outlet size", which may be expressed in terms of output Q and net head h by the parameter $L_o = Q^{\frac{1}{2}}/h^{\frac{1}{4}}$ which gives a value approximately proportional to the impeller diameter or other linear dimension at "outlet" in a series of pumps of homologous design.

(4) The head h in relation to the output Q and speed N which when combined in the form $NQ^{\frac{1}{2}}/h^{\frac{3}{4}}$ give a value which apart from the influence of scale effect would be sensibly constant for a range of geometrically similar pumps operating at constant head. This, of course, is the expression for specific speed N_s, which has little practical significance as a speed, but is a number which characterizes the pump type, or more accurately, the "outlet type", and is the commonly accepted basis of comparison.

(5) The total head H which affects the impeller hub diameter and is of particular importance in the case of multistage pumps.

In Fig. 3 an attempt has been made to depict efficiency values against size for the whole range of pump types from centrifugal-diffuser to propeller. The slope of the curves has been established on the basis of tests on many hundreds of pumps of one particular make in order to ensure homology, but the actual values

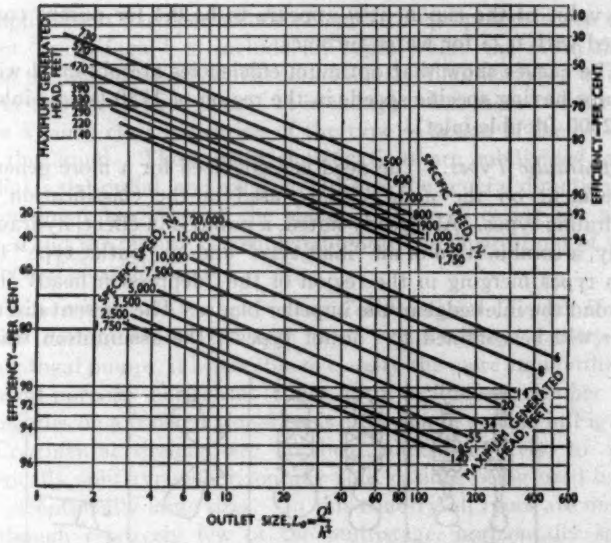


Fig. 3. Pump Efficiency in relation to Size and Specific Speed
Values are for single-inlet impellers. For double-inlet impellers total Q must be halved in evaluating L_o and N_s.

are representative of the best practice of the world's leading makers. While the main object in preparing these curves has been to indicate the effect on efficiency of variation in size and type, the values shown are such as might reasonably be expected from pumps of modern design operating under favorable conditions with a variation of, say, 3 or 4 per cent either way. In evaluating the expressions for size and specific speed, Q is in Imperial gal. per min. and is for a single-inlet impeller. For a double-inlet impeller total Q must be halved. The efficiencies are for single-stage pumps, but may be used for multistage pumps by making an appropriate reduction dependent mainly on the total head H.

The curves have been plotted with values of $(1 - e)$ as ordinates and in the formula devised by Moody for scale correction, viz.:—

$$\frac{1 - e_2}{1 - e_1} = \left(\frac{L_1}{L_2}\right)^n$$

the value of the exponent n appears to be 0.4 for pumps compared with 0.25 for water turbines.

The curves show that optimum efficiencies are obtained with pumps having specific speeds in the region of 1750 (single-inlet) or 2500 (double-inlet).

Hydraulic Types. The need is recognized for a more general agreement on the nomenclature used in the classification of hydraulic types. As already noted, a pump is in effect, hydraulically, a combination of an "inlet type" and an "outlet type" the two types merging in the region of the "depression head" just beyond the inlet edge of the impeller blades. The present discussion will be confined to "outlet types", the assumption being

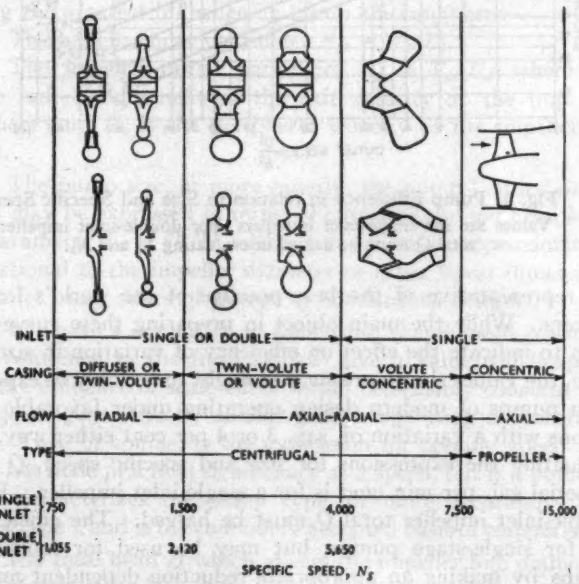


Fig. 4. Classification of Hydraulic Types

made that the "inlet type" has been selected to give a good efficiency rather than a relatively low value of "depression head".

Fig. 4 is a suggested classification of hydraulic types using the generally-accepted basis of specific speed as the type number.

Simplification has been aimed at, and it is proposed to use the term "centrifugal" to include all types where the mean flow from the impeller has a radial component. All other types would be in the "propeller" class. It is thought that these two terms give a fairly clear indication of the type of action of the blades on the liquid. These main classifications are subdivided into radial, axial-radial, and axial, to indicate the general direction of transitional flow through the blades. Types of casing are also given and the picture is completed by outline illustrations of the principal classifications.

Features of Construction. (*1a*) *Constructional Types: Horizontal Arrangement.* Although there exists a large variety of centrifugal pumps, it is possible to classify the more standardized general-purpose horizontal types into a limited number of categories on a constructional basis as shown in outline in Fig. 5.

Continental designs are confined almost entirely to the vertically split types, horizontally split casings being used only on exceptionally large sizes. In this country all types are made (although relatively few of the multistage, horizontally split type) the most appropriate design being selected to suit each application. At one time designers and users in America had an overwhelming preference for the horizontally split casing pump for all purposes; but with conditions becoming more onerous in certain fields, particularly higher pressures and temperatures in oil refineries and power stations, the vertically split casing type has become increasingly popular. This has usually taken the form of the barrel-casing construction into which is inserted either vertically split sectional stages or a single casing split horizontally.

In considering the relative merits of the vertically split sectional construction and the barrel casing construction for high pressures and high temperatures, two of the main factors to be taken care of in the design are:—

(1) To avoid excessive stress in the casing and main bolting with a sudden increase in temperature from minimum to maximum.

(2) To avoid interstage leakage with a sudden fall in temperature from maximum to minimum.

A correctly designed pump of the vertically split sectional type meets these requirements satisfactorily and boiler feed

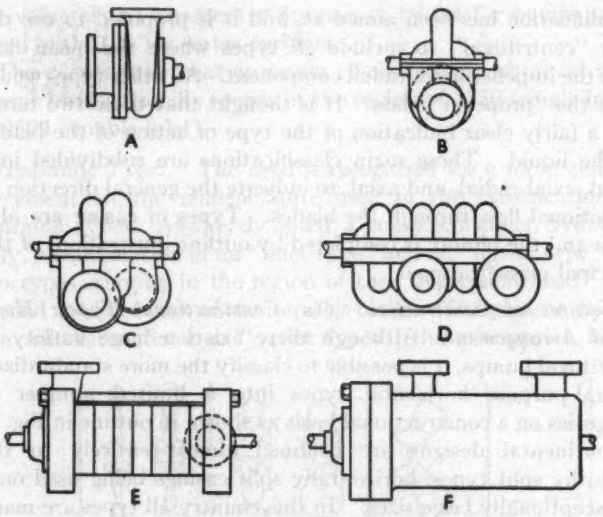


Fig. 5. Horizontal Centrifugal Pumps: Classification of Constructional Types

(A) Single-stage, vertically split casing of the volute, twin-volute, or diffuser type; single-inlet cantilevered impeller.
(B) Single-stage horizontally split casing of the volute, twin-volute, or diffuser type; double-inlet impeller mounted between two bearings.
(C) Two-stage horizontally split casing of the volute, twin-volute, or diffuser type; single-inlet or double-inlet impellers mounted between two bearings.
(D) Multistage, horizontally split casing, volute or diffuser type; single-inlet impellers.
(E) Multistage, vertically split, sectional-casing, diffuser type; single-inlet impellers.
(F) Multistage, barrel casing, diffuser type; single-inlet impellers.

pumps for pressures up to 3000 lb. per sq. in, and hot oil pumps for temperatures up to 650 deg. F. have been constructed on this principle. Essential features in design are: heat insulation between pump casing and main bolts so that the temperature of the bolts is always lower than the temperature of the casing; appropriately heavy section bolts and covers; correct proportioning of the area of the interstage body joints in relation to the cross-sectional area of the main bolts.

The cycle of bolt stress and inter-facial joint-pressure variation with temperature change is illustrated diagrammetically in Fig. 6.

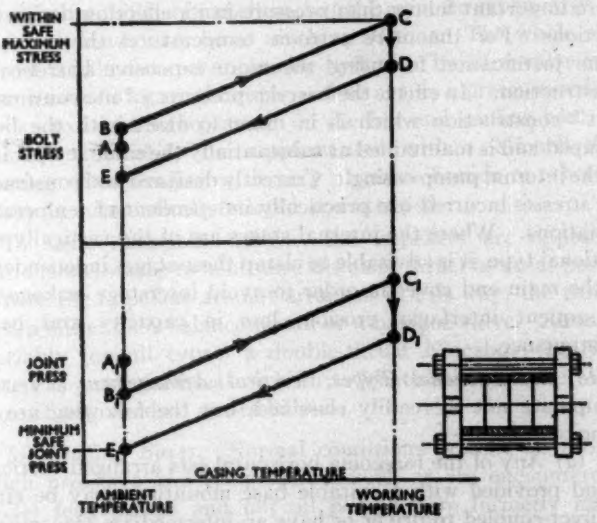


Fig. 6. Cycle of Bolt Stress and Joint Pressure Changes in High-Pressure, High-Temperature, Multistage Centrifugal Pump

A Pump bolts tight.
B Hydraulic pressure imposed.
C Working temperature suddenly applied to casing.
D Bolts gradually heated.
E Casing suddenly cooled.

Points AA_1 represent respectively the bolt stress and joint pressure after the initial stretching of the bolts. BB_1 is the corresponding condition with maximum hydraulic pressure in the pump. Sudden imposition of full temperature raises the bolt stress and joint pressure to CC_1 and the pump is designed for this extreme condition. If full temperature is maintained, the bolts take up a proportion of the full temperature and conditions are stabilized at DD_1. Sudden reduction of temperature to the minimum value reduces bolt stress and joint pressure to EE_1. Experience has shown that the residual pressure E_1 must

be between five and ten times the working hydraulic pressure depending on the type of joint, to ensure tightness under all conditions of operation.

It will be noted from the diagram that temperature is a far more important factor than pressure in its effect on design proportions. For the more extreme temperatures there may be some justification for using the more expensive barrel-casing construction. In effect, the barrel represents a "one-continuous-bolt" construction which is in direct contact with the liquid pumped and is maintained at substantially the same temperature as the internal pump casing. Correctly designed and constructed the stresses incurred are practically independent of temperature variations. Where the internal stages are of the vertically split sectional type, it is advisable to clamp these stages independently of the main end cover in order to avoid interstage leakage and consequent interfacial erosion, loss in capacity and heavy maintenance.

(*1b*) *Constructional Types: Vertical Arrangement.* Vertical pumps are not so readily classified, but the following are the principal types:—

(*a*) Any of the foregoing horizontal sets arranged vertically and provided with a suitable base mounting; may be either direct-coupled to driver or have an intermediate transmission shaft.

(*b*) Pump suspended below driver either on end of a long pipe, with intermediate shafting, e.g. borehole pumps; or close-coupled to the driver.

(*c*) Pumps with direct-coupled submerged-motor drive.

Vertical pumps show an advantage over horizontal units in economy of floor space and foundations and for installations where there is a wide variation in water level on the supply side.

(*2*) *Hydraulic Thrust.* Radial thrust due to unsymmetrical discharge conditions around the impeller may cause serious lateral deflexion of the pump rotor when developing a high head, particularly when the pump is of the single-stage type with the impeller cantilevered from the bearings. This trouble is liable to occur when the casing is of the simple volute type. It can be alleviated by using a twin-volute casing or a multi-port diffuser.

Axial thrust is an important factor in multistage pumps as it is customary to use impellers of the single-inlet unbalanced type

in order to avoid excessive span between bearings. The standard practice in this country, is to arrange all the impellers facing in the same direction; this has the advantages of simplicity of design and interchangeability of impellers, but the axial thrust on large sizes and high heads may be as much as 50 tons. The whole of the thrust is taken on an automatic hydraulic balance valve which also serves to locate the rotor and to act as a pressure-reducing device to avoid high pressure on the free-end stuffing box. The balance valve may be supplemented with a Michell thrust bearing when dealing with very abrasive liquids or where the pump is automatically started and stopped.

In some American designs, the impellers are opposed in various ways so as to neutralize the axial thrust as far as possible. Where the impellers are not arranged in this way, the thrust is taken either on a balance piston or a balance valve. Almost invariably (on all types) a double-thrust Kingsbury bearing is fitted as a standard feature both to locate the rotor and to take care of any residual out-of-balance.

(*3*) *Stuffing Boxes.* Normal conditions present no problem. High pressures and temperatures such as are encountered on boiler feed pumps and hot oil pumps were formerly met by using a sealing liquid in conjunction with a leak-in and leak-off bush which ensured that the stuffing box packing only came in contact with low-pressure, low-temperature liquid. Modern practice has in many instances dispensed with the need for this rather complicated (though very effective) arrangement. High temperature is dealt with by intense precooling of the drip before it reaches the packing. Where packing is subjected to hydraulic pressure, the compressive load is concentrated on one or two rings adjacent to the gland and experience has shown that providing the design is such that extrusion of the packing under the gland is obviated, then remarkably high pressures and rubbing speeds can be met satisfactorily without excessive heating and consequent sleeve wear, even when ordinary soft packing is used.

A sealing liquid may still be required where the liquid pumped has no lubricating properties or is very abrasive, unstable, or toxic. However, on many applications such as chemical process work and oil refining, particularly where catalysts are used, objections are sometimes raised to dilution or contamination of the

product, and this has led to the wider use of mechanical seals in recent years. In this design, a seal is formed between two radial faces, one stationary, one rotating. Single seals are used where conditions permit, but double seals may be required for the more difficult conditions and for the latter a sealing liquid is introduced between the two seals for lubricating and cooling the operating faces.

A more recent development for difficult conditions, is the elimination of the stuffing box by utilizing the submerged motor drive which has been applied successfully to borehole pumps for many years. Promising new fields are for pumping high temperature water at saturation pressure, certain corrosive liquors and light hydrocarbons. In the latter connection it may be mentioned that designs are under consideration for light hydrocarbon pumps to develop a pressure of 18,000 lb. per sq. in.

(*4*) *Materials.* "Fluids" pumped range from volatile liquids to drinking water, from heavy abrasive slurries to solids and fibres, from the supreme corrosive hydrofluoric acid to one of the heaviest metals—molten lead. Temperatures may be anything up to 900 deg. F. and working pressures higher than 3000 lb. per sq. in. To meet these widely varying conditions, pumps are made (or lined with) almost every kind of material, selected according to service, and in addition to the more frequently used materials such as cast-iron, steel, bronze, and their many alloys (in particular the austenitic stainless steels), they include lead, stoneware, rubber, glass, synthetic resin, etc.

Interesting Examples. Centrifugal pumps may deliver three or four gallons a minute, or millions of tons a day. Working heads may be nearly two miles or a few inches. They may draw from a vacuum exceeding 29 inches of mercury, or have an inlet pressure of 1000 lb. per sq. in. Temperatures may range from minus 300 deg. F. up to 900 deg. F. The liquid may be acid, alkali, or neutral; clear or abrasive; thin or very viscous; lubricating or non-lubricating; unstable or toxic; or may combine a number of these properties. One of the highest head pumps in the world handles hot oil and develops 8600 feet in a single casing. Propeller type pumps for flood prevention discharge over 5,000,000 tons a day each against a head of 10 feet or less with a propeller 12 feet in diameter; boiler feed pumps have been con-

structed for a discharge pressure of 3000 lb. per sq. in. requiring over 3000 Bhp. to drive. The largest vertical volute-type pumps ever built operate in conjunction with a hydro-electric power installation and are for irrigation service: the net head is 295 feet, requiring an impeller nearly 15 feet in diameter, and each pump is driven by a 65,000 Bhp. motor. Under certain conditions the pump may operate as a water-turbine and generate electric power. Another interesting hydro-electric scheme under construction employs pumps for lifting the water from a river over a mountain divide to drive turbines on the other side. These pumps will have 144-inch diameter discharge branches and will each deliver over 500,000 gal. per min. Of exceptional merit, on account of the high head in a single stage for such a large output, are the 75,000 gal. per min. aqueduct pumps developing a head of 476 feet and powered by 12,500 Bhp. motors. Probably the most outstanding example of a combination of high head and high speed in a single stage are crude oil pumps for a duty of nearly 6000 gal. per min. and 900 feet at 3600 Rpm. For land drainage, double-suction volute pumps up to 1000 tons a minute have been installed. Pumps of 36,000 Bhp. each and for heads up to 1000 feet have been supplied for pumped storage in connection with hydro-electric plant.

Speculative Survey of Future Developments.

In this country a "water grid" is probably outside the realm of practical economics on account of the diversity in supply sources and consumer requirements, apart from the difficulties and expense in superimposing an entirely new system on the existing complicated network of supplies. We are likely to see an extension of regional groupings with the ultimate object of ensuring that every domestic and industrial consumer has a piped supply. Appropriation of existing supplies of suitable elevated storage and river water will necessitate extensive resort to the utilization of underground storage—particularly in the south. There is little doubt that the centrifugal borehole pump with surface motor and intermediate transmission shaft will be the principal type used for dealing with this underground supply, but for the smaller and intermediate capacities we are likely to see an extension of the use of the submerged motor-pump

mainly on account of its lower first cost and its greater flexibility and adaptability. Pumps for dealing with surface supplies will continue to be of the high efficiency type.

Centrifugal and propeller pumps will make a fundamental contribution towards developing and raising the standard of life in such backward areas as India, China, Kenya, and the Near and Middle East, by their use on a wide scale for irrigation. In India the short-term plan to install thousands of centrifugal borehole pumps distributed over the area to be cultivated will later be replaced in the north by the melting snows of the Himalayas, requiring large capacity centrifugal pumps for boosting. In Egypt and the Sudan there will be a growing demand for large capacity propeller pumps lifting the water a few feet from the Nile to the irrigation canals.

The 1939-45 war provided notable examples of the use of centrifugal pumps for boosting large volumes of oil along pipelines.

Important developments of the future include two pipelines, from the Persian Gulf to the Mediterranean, of 30 or more inches diameter, each of which will have a capacity of 15 or more million tons per year. High-pressure boosting stations will be required and will be equipped with centrifugal pumps.

Nationalization of the mines in this country may lead to regional grouping of collieries and compounding of drainage requiring pumps of larger capacity. New mines will be deeper, calling for higher head pumps and increased head per stage to obviate series arrangements. There should be a growing world demand for land drainage pumps particularly of the propeller type.

In this country feed pumps for around 800 and 1250 lb. per sq. in. pressure are likely to constitute the main demand in central power stations, although in a few instances pumps for 1700 lb. per sq. in. pressure may be required. Similar conditions are likely to prevail in the U.S.A. although, due to the popularity of superimposed topping units, the proportion of high-pressure stations may be higher. On the continent a wider variety of boiler types is employed and feed pressures will range up to 3000 lb. per sq. in. as at present.

During the initial period following the introduction of high-

pressure power stations in this country, the feed pumps were placed between the final heaters and the boilers, and these pumps dealt with feed temperatures up to 420 deg. F. without difficulty. With the development of the high-pressure heater the need for high-temperature feed pumps disappeared and in general temperatures are not likely to exceed 250 deg. F. in the future. Simplification of plant by omission of variable-speed devices from feed pumps is likely to be maintained, although by correct incorporation in the heat balance there may be a tendency to extend the use of steam turbine driven pumps as main operating units. The successful application of the feed-circulating pump to boilers of the La Mont type may lead to the extension of the forced-circulation principle to boilers of the orthodox water-tube type, for capacity increase or peak boosts. There is a growing interest in feed-circulating pumps for chemical dope conservation. Pumps for these services may be of the submerged-motor type.

Increasing use of syphon recovery in power stations of the non-cooling-tower type will develop the need for condenser cooling-water circulating pumps of the high specific-speed type with steep head-gallons characteristic.

The eventual application of atomic energy to electric power production, if used as heat for the generation of steam, might involve little or no change in the use of centrifugal pumps, but, if used directly in gas turbines, the need for feed pumps, extraction pumps, etc. would diminish.

The development of district heating schemes both here and abroad will require large-capacity hot-water circulating pumps at moderate pressures.

The increasing tempo of the oil age will require the erection of new refineries in various parts of the world and the growing complexities of the chemical-hydrocarbon reactions involved will call for greater refinements in centrifugal process pumps. In the same category are process pumps for the chemical industry which continues to expand particularly in the development of new petroleum-based products.

In the paper-making industry, stock agitation by pumping will replace the less efficient mechanical agitators. The use of pumped circulation in the caustic digestion of straw and esparto halves the time required. Water at 1400 lb. per sq. in. used for "barking" may be developed for "splitting".

Most existing hydraulically operated plant in steel mills, etc., using centrifugal pumps employs a pump pressure of about 750 lb. per sq. in. The present tendency is to increase this pressure to 1500 lb. per sq. in.

If current schemes materialize for water power development in Scotland, large high-capacity, high-head pumps will be required for peak load storage.

Probable Trends in Design.

Future trends and developments in the art of pumping are likely to be mainly in the direction of a continued widening of the fields of application already outlined. This will provide unprecedented opportunities for standardization and large-scale production giving scope for greater type specialization, the development of new and improved types, and the introduction of novel methods of construction.

Among probable trends in pump design may be noted:—

(1) For general purposes and for small and medium capacities and heads, a tendency to adopt the simple and compact combination pump-motor unit with impeller mounted on the motor-shaft extension. May be used for horizontal or vertical arrangement.

(2) An increasing adoption of the vertical arrangement for single-stage single-inlet pumps of the volute type for water supply, irrigation, etc.

(3) The use of borehole type pumps, single-stage and multistage with short suspension pipe, for booster service, waterworks, condensers, drainage, irrigation, fire service, etc. Efficiencies as high as horizontal pumps are now possible and the main advantages are flexibility, small foundations, self-priming at starting, and safety of motor.

(4) Increased adoption of twin-volute diffusers giving symmetrical discharge from impeller and balanced radial forces.

(5) Increased speeds by using steam turbine, gas turbine, or higher frequencies, permitting use of single-stage for higher pressures.

(6) Higher heads per stage for multistage pumps to enable the number of stages to be reduced, particularly for the highest pressures, as for boiler feeding and hot oil service. The difficulty

may be to find suitable materials to withstand erosion at the high peripheral speeds.

(7) Extended use of medium and large propeller pumps for flood control, irrigation, and condenser circulation. Variable-pitch propellers may be more widely adopted.

(8) Employment of the newer and more expensive alloys in the construction of process pumps for the chemical and oil refinery industries. Experience has shown the advantages of installing the highest quality pumps which although higher in first cost involve lower maintenance and fewer service interruptions.

(9) A tendency for paper stock pumps to increase in size to keep pace with the increasing speeds of paper-making machines. The increasing scarcity of spruce will call for materials to resist the corrosive effects of the bleaching agencies required by other types of wood.

(10) Extended use of mechanical seals, replacing the orthodox stuffing box particularly on process pumps.

(11) Development of the submerged motor drive for pumps dealing with high-temperature water, hydrocarbons, acids, and other liquids where stuffing boxes are undesirable.

(12) Vertically split casings replacing horizontally split casings for many process duties.

(13) Increasing popularity of the self-priming pump for small units up to about 25 Bhp. Efficiencies are now almost equal to those for straight centrifugals.

(14) For household water supply a decided trend towards the adoption of the jet-centifugal self-priming pump.

PRACTICAL ENGINE ROOM TESTS OF LUBRICATING OIL CONTAMINANTS.

This article by F. L. Gerin is reprinted from "Motorship" for June, 1948. As bearing loadings are increased in new designs freedom from contaminants in oil will become more important.

Tests to measure critical contaminants in lubricating oils are not difficult and several benefits accrue when they are made in the engine room by the operators of the engine. This article

FIGURE 1.—The Testing Equipment is Stored in a Compact Cabinet as Shown in the Phantomed View Above. A Special Chest Requiring Less Headroom Has Been Designed for Marine Use.

will describe simple means for measuring, in lubricating oil, the entire group of contaminants considered responsible for all engine deposits as well as water, dirt and metal particles, acidity, and viscosity—showing fuel dilution. It is also possible to determine the water and sediment in fuel, and this entire process is within the skill of an engine room operator.

When these tests are performed by the engine operators, they put lubrication on a level with the other vital operating conditions regularly observed and recorded in the log. The importance of knowing that the lubricating oil is in proper condition is well understood and is reason enough for oil inspections.

But there is another reason, equally valuable although not so well known, for making tests each week. Changes in contaminants and in the rate of contamination are a sensitive indicator of altering conditions inside the engine. Regular oil inspections can show up many derangements early enough to prevent more serious symptoms. Stuck rings and plugged oil rings, for instance, reflect in an increased rate of gummy contamination; partly clogged spray nozzles cause fuel dilution, as do leaky fuel lines; failure of water jacket seals can be predicted by the first discovery of water; scuff metal particles means possible seizure conditions, etc. And every so often, scrapping the entire charge of lubricating oil in the engine is avoided by discovering the leakage of water or fuel before it has gone too far.

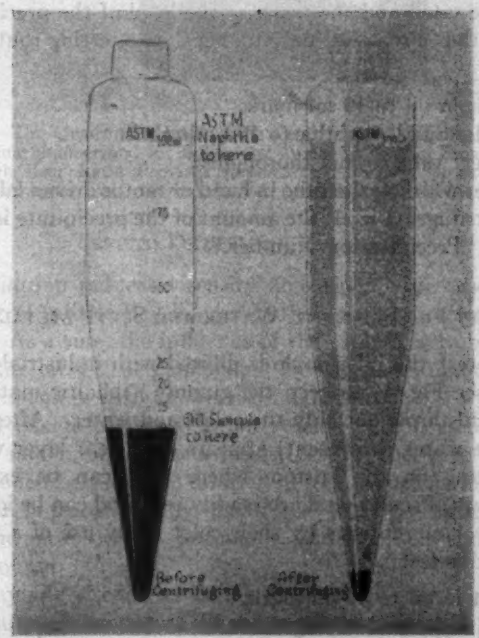


FIGURE 2.—The Cone Shaped Tubes Used to Determine the Precipitation Number. Top of the Dark Portion in Tube at Right, Read Against the Graduations, Gives Precipitation Number.

A continuous record on the oil offers to engineers responsible for successful operation the peace of mind that attends knowing what is going on inside the engines, the ability to furnish running curves and data to management and when desired, to engine builders as proof of proper lubrication.

Test to Measure Deposit Forming Substances.

Asphalts, resins and gums are considered responsible for all engine deposits because they are the binder material for the dirt, free carbon and metal, and they are strongly attracted to metal surfaces. The entire group of these oil-breakdown substances precipitate out of the oil sample when it is diluted with special naphtha. The cone shaped tube (Fig. 2) used for making the test is graduated and the mark at the level of the precipitate is known as the "Precipitation Number." The testing routine is as follows:

(1) Pour in oil to 10 ml mark.
(2) Add ASTM naphtha to 100 ml mark.
(3) Invert tube to mix thoroughly.
(4) After whirling the tube in hand or motor driven laboratory centrifuge, (Fig. 3) the amount of the precipitate is read as the "Precipitation Number."

Test For Sediment, Water and Scuff Metal.

In this test the oil sample is diluted with industrial 90 deg. benzene (see Fig. 4) to keep the gummy asphaltic materials in solution and throw out only the solids and water. After centrifuging, the water (if present) appears as a clear layer with the solids resting on the bottom where they can be examined. Bright flakes of scuff metal are readily seen and can be identified as ferrous or non-ferrous by their color or by use of a magnet. To make this test:

(1) Pour in oil to 50 ml mark.
(2) Add industrial 90 deg. benzene to 100 ml mark.
(3) Mix thoroughly and centrifuge.
(4) Record amount of water and amount of solids as water and sediment; examine nature of the sediment.

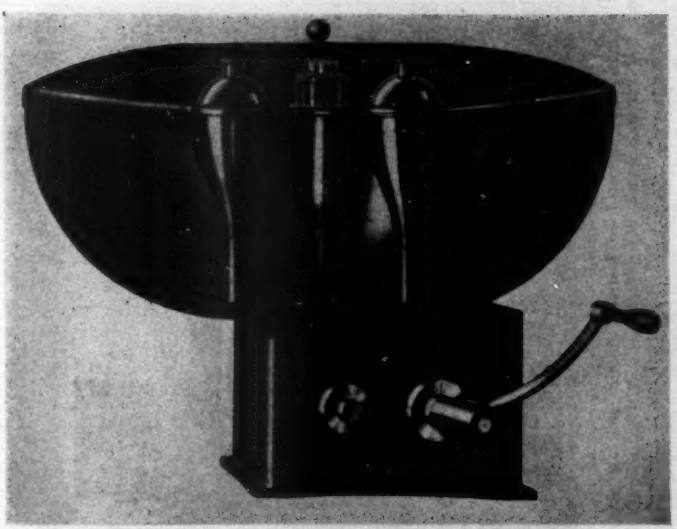


FIGURE 3.—The Quadruple Head of the Centrifuge Allows Tests for Precipitation Number and for Solids and Water to be Run Simultaneously. A High Gear Ratio Provides Reasonably Effortless Operation of the Hand Powered Unit. Power Driven Models Are Also Made.

TEST TO DETERMINE VISCOSITY.

Fuel dilution reduces viscosity and asphalt increases it, but the effect of a little fuel is greater than that of a great deal of asphalt. As a rule, the influence of the asphalt is to be ignored because the oil will be condemned by the centrifuge test when there is enough of it to be taken into account.

The ill effects of fuel dilution recently are becoming more generally understood. The viscosity test for fuel is to be highly recommended because fuel dilution in lube oil occurs much more often than is recognized by operators. In some cases, it comes slowly and more or less continuous, and in others suddenly, in large amounts.

The Saybolt viscometer is the most widely used instrument for measuring viscosity. When an oil is rated, for instance, "95 seconds Saybolt at 100 deg. F.," it means that when tested at a temperature of 100 deg. F., the oil fills the receiving flask in 95 seconds of time. Viscosity measured in other instruments,

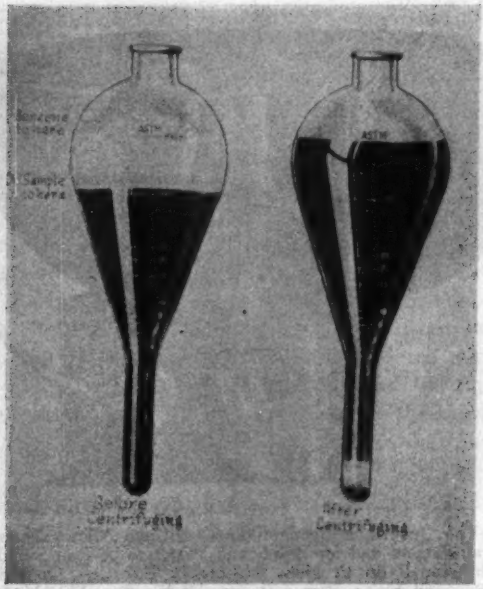


FIGURE 4.—The Tubes Used to Test for Solids and Water. The Solids are Shown in the Bottom of the Right Hand Tube with the Light Area Above Representing a Layer of Water.

for example, Kinematic, can be translated to Saybolt and vice versa from simple conversion tables. The instrument illustrated in Fig. 5 has the official Saybolt dimensions. However it has a modification which shortens the time for making the test, and is the feature which makes the instrument quite practical for engine room use because it eliminates the need for skillful (laboratory level) handling. In the standard Saybolt, the tube and orifice are in one piece, whereas in this instrument they are not. In the Saybolt, the bath is first brought to temperature; the sample is heated separately in a pan and then poured into the viscosity tube. While the instrument illustrated can be operated in exactly the same way, the engine room operator is not so limited because he can provide more oil for test sample than is usually sent to the laboratory analyst. Therefore, in engine room use, the oil to be tested is poured directly into the bath with the viscosity tube

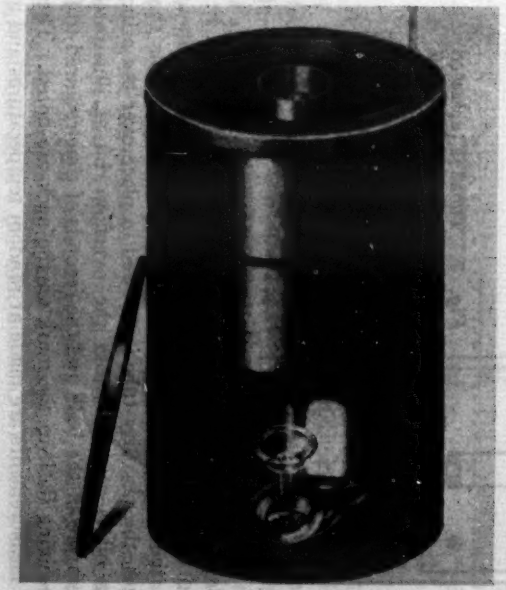


FIGURE 5.—Viscosity is Checked in This Viscometer of New Design. Sectional Views are Shown Below.

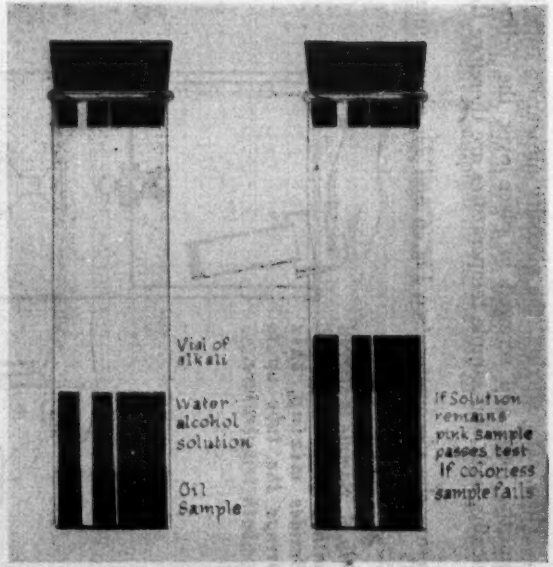

Vial of alkali
Water alcohol solution
Oil Sample
If Solution remains pink Sample passes test If colorless sample fails

FIGURE 9.—The Simple Method of Checking for Acidity is Shown. After Shaking, the Oil Rises, Leaving the Test Solution at the Bottom.

removed. Then, after stirring, the viscosity measuring tube is filled from the bath and screwed in place over the orifice. Automatically, bath and sample temperatures are therefore the same.

To operate the viscometer (see Fig. 6 and 7), a one quart

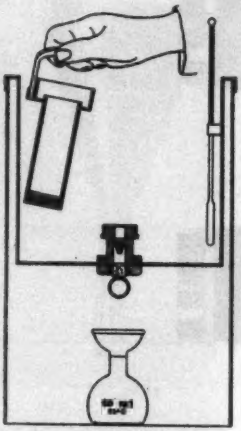


Fig. 6. Sample is stirred with the viscosity tube. Besides mixing, this brings the tube up to bath temperature.

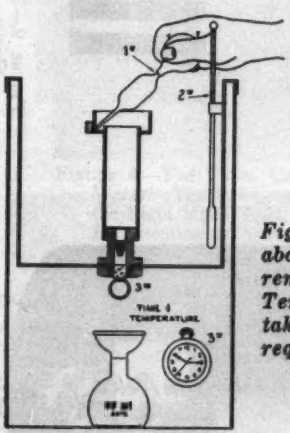


Fig. 7. Tube is screwed above orifice and excess oil removed from overflow rim. Temperature of bath is taken, cork pulled, and time required to fill flask recorded.

sample of oil at any temperature between 100 and 200 deg. F. is poured through 100-mesh screen supplied with the viscometer, filling the bath to the mark. After stirring for two or three minutes, the viscosity tube is screwed in place over the orifice. Excess oil is removed from the overflow rim as illustrated; temperature of the bath and the seconds of time required to fill the receiving flask to the mark on the neck are noted. These quick measurements are made at random temperatures and are

compared to previous readings made at other temperatures by reference to the A.S.T.M. Viscosity-Temperature Charts (Fig. 8) which are supplied by the makers of the viscometer.

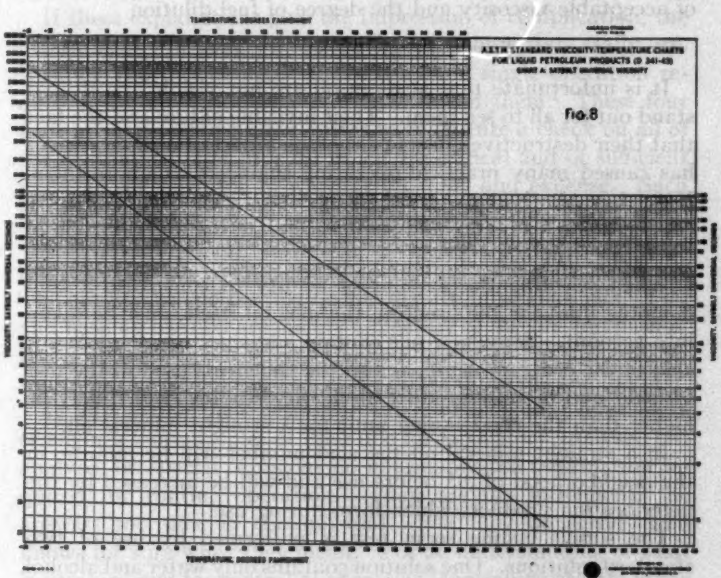


Fig. 8. Two typical viscosity reference lines are shown on a standard ASTM viscosity temperature chart. It can be seen how, once a line is established at random temperatures, viscosity at the standard test temperatures can be determined.

When the viscosities for random temperatures are plotted on the chart, a straight line drawn through them makes a line of reference to read the viscosity for any other temperatures. Thus the operator's readings are readily converted to viscosities for standard temperatures, *i.e.* 100 deg., 130 deg., or 210 deg.

These excellent charts serve both the diesel operator's purpose and his desire to save time. He can warm up a quart of the new oil to about 200 deg., and read the viscosity at several temperatures while the oil is cooling. This will establish several points for the base line. Other lines can be drawn to represent diluted oil, by determining the viscosity when fuel is added to the new lube oil in measured amounts. One ounce of fuel added to one quart of lube oil is 3 per cent dilution, 1.7 ounces make 5 per

cent dilution, 3.5 ounces—10 per cent, and 5.6 ounces—15 per cent. Thereafter, the operator can refer any readings on used oil to this chart and determine at once whether it is within a zone of acceptable viscosity and the degree of fuel dilution.

Test to Determine Acidity.

It is unfortunate that acids in oil are not visible and do not stand out for all to see them. Their invisible nature and the fact that their destructive effects pass off as a part of ordinary wear, has caused many practical operating engineers to ignore them. Yet, they slowly etch polished surfaces, some acids dissolve copper, some lead, etc., and some accelerate the deterioration of the lubricating oil.

When selecting the amount of acidity to constitute a condemning limit for the used oil, the new oil itself must be taken into account. Some new oils contain additives which while not acid in the ordinary sense, act on the chemicals used in the test for acid and thereby produce an acidity reading, which may be higher than the condemning limit for a straight mineral oil.

In a straight mineral oil the acid number will be minute, in the neighborhood of .05 before use and will be increased by the deterioration products which form during use.

The simple engine room test is performed by use of two standard solutions. One solution contains only water and alcohol with a touch of phenolphthalein indicator. This indicator is pink in the presence of alkali and colorless when neutral or acid. The second solution of water and alkali comes in vials each containing a simple dose of alkali (enough to neutralize acids in oil having an A.S.T.M. acid number .30).

To make this test, referring to Fig. 9, the used oil sample is poured into the glass cylinder to the lower mark. The water and alcohol with pink indicator is poured in to the next mark and one vial of alkali then added. Upon shaking vigorously, the water-alcohol will wash the acids out of the oil sample and they will react with the alkali. On standing for a few minutes, the oil will rise and if the water-alcohol layer is still pink, the acidity of the oil is below neutralization number .30 mg. of KOH as it is commonly expressed, but if the layer is not pink, the oil is considered to have failed the test. The value of .3 for used oil is considered conservative as a condemning limit.

Those who wish to use limits of more than .30 can start with three quarters of the normal sample of oil and one vial of alkali for acidity up to .40 and obviously acidity of .6 can be checked by doubling the dose of alkali, through pouring in two vials.

If these explanations give the impression of complication, the illustrations will show how utterly simple these tests really are. Reading the thermometer is probably the single operation requiring the most skill in performing any of them. These four tests are considered by the writer to constitute a check on all of the contaminants of lube oil which are critical and of sufficient value to the operator to warrant the time and expense. Such tests as "gravity," "fire" and "flash" are of no benefit when applied to used oil and usually of no benefit on new oil for that matter. Carbon residue and the like are in the same category. The gravity has no bearing on the quality of modern oils. Fire and flash on used oil are merely a measure of the fire and flash of traces of fuel which may be contained, whereas what the operator wants to know is not the flash of the dilutant but the amount of it.

In advocating these tests, the writer realizes that most users of engines send samples of their oils for laboratory analysis, but analyses on special occasions do not supply the information required for routine operation. They are too infrequent, the answers too slow in coming, and when they do arrive, who knows for sure what they mean? For an understanding of what is going on and for control of operation, continuity of data is the thing.

WHAT THE USER SHOULD KNOW ABOUT TUBING.

This article by E. J. DeWitt, describing the principal classifications, characteristics, and fabrication of tubes, is reprinted from "Machinery" for February, 1948.

The uses for tubing in product design and manufacture are almost unlimited. It can be applied as a structural member to support machine parts or it can be used to carry gas, air, steam, water, oil, or other fluids. In many cases, it can be employed for dual purposes—as a structural part and also to transport a fluid.

For each of these uses, the correct type and class of tubing must be specified, or else the application will result in failure. Such a choice is not simple, as there are many types, sizes, and classes available. It is only by having a knowledge of the characteristics of each and the method by which it is manufactured that the proper type can be chosen.

Classifications of Tubing.

The various kinds of tubing available can be divided into two general types—tubes with welds or seams and tubes without welds or seams. Each of these types are subdivided into three classifications, namely, pipe, mechanical tubing, and pressure tubing. Alloy tubing and oil country goods are further divisions of these three classes.

There may be some question as to the inclusion of pipe in a list of tubing, but broadly speaking, pipe is tubing made to a certain standard; it is generally designated by the nominal inside diameter. It has a fixed wall thickness designated at the mill and by engineers as a schedule number, and referred to by tradesmen as standard, extra heavy, and double extra heavy. The nominal pipe sizes, weights per foot, and sizes of couplings for the various schedules can be found by reference to the catalogues of tube manufacturers.

Mechanical tubing, as its name implies, is designed and manufactured with mechanical uses in mind. Tolerances are closer than on any other common class of tubing. These closer limits save much time when the tube is to be machined, and make chucking in modern automatic screw machines feasible from the viewpoint of economy. Mechanical tubes are available in a wide range of sizes; before the War Production Board reduced the number of sizes to 528, some mills and warehouses listed between 1500 and 2000. Another advantage of mechanical tubing is that many grades are partially annealed to improve machinability after cold-drawing. The result is a tube with sufficient hardness to be machined without tearing.

Pressure tubing differs from mechanical tubing in many respects. First, being primarily intended for pressure uses, it is pressure tested; second, the dimensional tolerances are less restric-

tive; and third, most pressure tubes are fully annealed, since many applications involve working of the tube. For example, heat exchangers are rolled into tube sheets and boiler tubes are bent at sharp angles.

Alloy tubing is a subdivision of the classifications previously mentioned. It may be either alloy pipe, alloy mechanical tubing, or alloy pressure tubing. One of the best known of these is an alloy tube known as the S A E 52100 series. It is an alloy mechanical tube of highest quality and made to closer tolerances than most other mechanical tubes; it is the basic tube for bearing race manufacture. There is also the well known S A E 4130 series. This series of mechanical tubes gained wide usage during the war. Airplane parts, such as motor mounts, landing wheel supports, and other members were made from it. It is a mechanical alloy tube which is usually supplied in a normalized condition and can be heat-treated to produce great strength after it has been fabricated or welded. The aircraft engine mount shown in Fig. 1 is constructed of S A E 410 material.

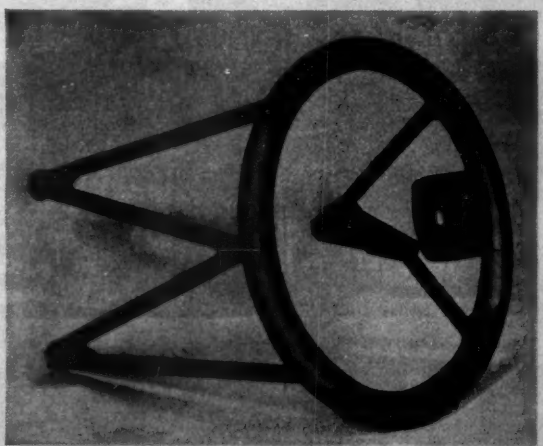


FIGURE 1.—A Typical Structural Application of Tubing is the Aircraft Engine Mount Illustrated.

Stainless tube is another alloy tubing of considerable importance. It is also available in three classifications—stainless pipe, stainless mechanical tubing, or stainless pressure tubing. In most cases, it can be obtained in welded or in seamless types.

"Oil country goods" is a broad term covering a not too generally known but none the less large portion of all tubular products. Examples of oil country goods are pipe lines, oil-well casings, drill pipes, and others.

The Manufacture of Welded Pipe.

Pipe is the cheapest form of tubing suitable for carrying a fluid or a gas, and is available either in the welded or seamless type. Four generally used processes are employed for making welded pipe: (1) Lap welding; (2) gas welding; (3) electric resistance welding; and (4) continuous butt welding.

The first process—lap welding—requires a preheating furnace long enough to handle strips of steel that are longer than the finished length of pipe desired. The furnace must also be wide enough to keep an adequate number of strips hot ahead of the welding unit. (See Fig. 2.)

The method of welding is the same as that used by the blacksmith. The two edges are lapped, and while at welding temperature, are squeezed or hammered. In making lap-welded pipe, the

Figure 2.—Loading a Preheating Furnace for a Lap-Welding Mill at the Sparrows Point, Md., Plant of the Bethlehem Steel Co. After Being Heated, the Strip Stock is Formed into a Rough Tubular Shape in a Bending Furnace.

strip, or skelp, is heated and then formed by rolls to a rough tubular shape in a bending furnace.

The "pipe round," as the product of the latter operation is called, is reheated and welded in the welding rolls, which can be seen at the right of Fig. 3. The heavy drive mechanism used to

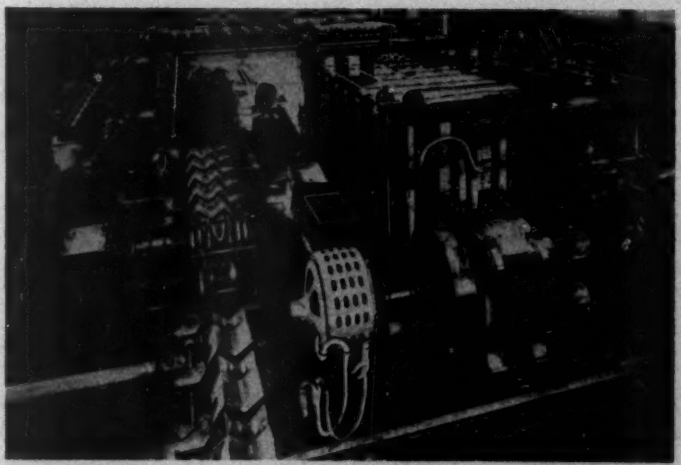


FIGURE 3.—General Arrangement of the Bethlehem Lap-Weld Mill. The Pipe is Welded in Rolls at the Far Right of the Illustration, Straightened in the Machine Shown at the Left of the Furnace, and Finally Cut, Cooled, and Inspected.

operate these rolls is indicated at the left of the drive rolls. The incandescent pipe shoots out of the welding rolls into a trough, and then rolls down the rail shown at the front of the drive unit into the conveyor at the left of the motor. This conveyor leads through the first sizing roll, which is shown at the end of the conveyor, and then into the straightening machine located just left of the center of the photograph. From there it passes to the end of the line where it is cut, cooled, and inspected.

Lap welding, which is by far the oldest method of welding pipe, is rapidly becoming obsolete. Advances in other methods of manufacturing have now made it possible to produce pipe of like quality at a lower cost. Lap welding is limited to pipe sizes of 2 inches and over.

Gas welding requires much simpler equipment, and is the cheapest process to install. A strip, usually rolled in cold form, is placed on a reel at the end of a set of forming rolls. As the strip is drawn through the forming rolls, its contour is changed from a flat to a circular shape as illustrated in Fig. 4. When the shape is fully round, the joint passes under a row of preheating gas torches that gradually build up the temperature of the joint to nearly the welding temperature.

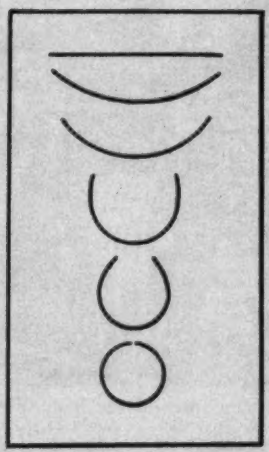


Fig. 4. A Series of Rolls is Used to Form Strip Stock into a Cylindrical Shape, as Shown. The Pipe is then Welded with Torches or Electrodes

A pressure roll, usually at right angles to the forming rolls, then squeezes the edges together; while the joint is under pressure, it is passed beneath an oxy-acetylene welding torch. A second pressure roll may be used in some processes to squeeze the weld area while a stream of cold water flows over the pipe to quench it.

Straightening rolls are usually incorporated in the line of pulling rolls, and an automatic cutoff device completes the unit. The automatic cutoff machine ordinarily is set to produce lengths of 21 feet to limits of plus or minus 1 inch.

Electric resistance welding, which is being increasingly used for smaller installations, is essentially the same as the gas welding process. The major difference is the use of a pair of water-cooled copper rolls as electrodes, which are so designed that a current of welding intensity is delivered to the edge of the tube. Pressure rolls restrain the soft edges until the weld area is quenched, and straightening and cutting operations complete the pipe. The discharge end of a forming and welding machine is shown in Fig. 5.

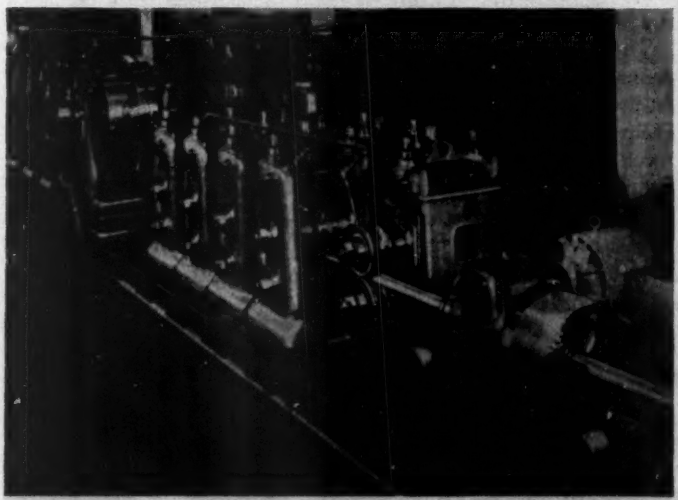


FIGURE 5.—The Discharge End of a Cold-Forming and Electric Welding Machine Made by the American Electric Fusion Co. The Large Rolls at the Left are the Welding Electrodes; at the Far Right is the Cut-off Machine.

The continuous butt welding process, while costly to install, is the most economical of the pipe welding methods on account of the high production possible. The tube mill of the Bethlehem Steel Co. at Sparrows Point, Md., is shown in the heading illustration. The range of pipe produced on this type of mill is generally from ½ to 3 or 4 inches in diameter, and the production is about 30,000 feet per hour of ½ inch, Schedule 40 pipe.

The continuous welding process follows the general pattern of

both gas and electric welding. Skelp is welded continuously end to end by an automatic butt-welding machine, and once welded, is allowed to loop itself on the floor before being fed into the furnace. It is this looping arrangement that makes continuous welding possible, for it gives time to butt-weld one coil of skelp to another without having to stop the flow of stock through the furnace. The skelp is brought up almost to the welding temperature by the long gas-fired furnace. Just outside the furnace, on the exit side, a pair of rolls is located. These rolls change the shape of the skelp from flat to round. After the skelp has been rounded, it moves to the weld area.

The furnace does not heat the strip to the welding temperature. This is done by playing a stream of air or oxygen-rich air on the two edges of the skelp. The reaction of the air on the hot skelp is identical to the reaction of oxygen used in a cutting torch. The hot steel in the area affected becomes fluid and is squeezed in pressure rolls to form a perfect butt weld while in this nearly liquid condition.

Visualize the skelp as it travels through the furnace and rolls at 500 feet per minute. This is the equivalent of approximately

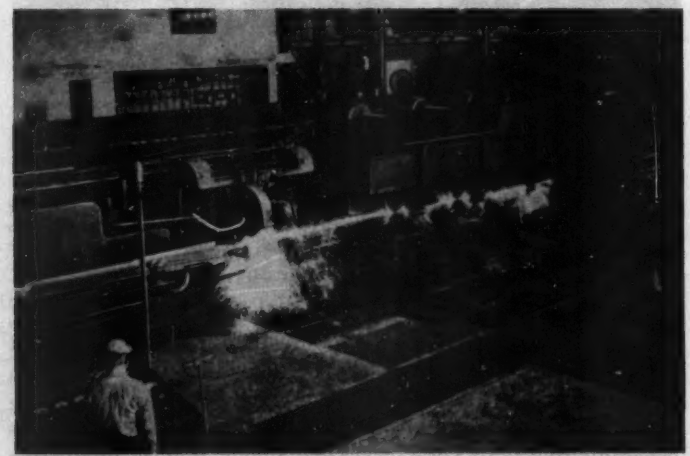


FIGURE 6.—The Automatic Cut-Off Saw at the Discharge End of a Continuous Tube Mill Makes Twenty-Five or More Cuts per Minute on Pipe Traveling at the Rate of 8 Feet per Second.

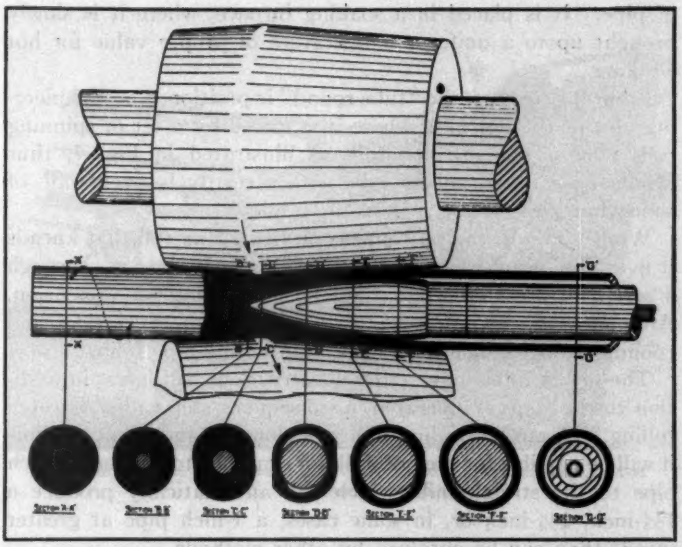


FIGURE 7.—Cross-Sectional Drawing Showing Diagrammatically the Result of the Piercing Operation in the Manufacture of Seamless-Steel Tubes—Courtesy National Tube Co.

8 feet per second. In about one-third second, the pipe is changed from a flat strip into a welded pipe.

The production of pipe at these high speeds is dependent upon the efficiency of the cut-off mechanism. Fig. 6 shows the automatic cutting-off machine in operation. Its ability to make twenty-five or more cuts per minute while the pipe is traveling at 8 feet or more per second is one of the major factors contributing to the present low cost of pipe.

THE MANUFACTURE OF SEAMLESS PIPE.

Several different processes are used to make seamless pipe and tube, but they are sufficiently similar so that if one is described, the reader will understand them all. Whereas the processes described for making pipes with seams begin with a strip, the seamless pipe is made from a solid bar or "tube round." The "tube round" of carefully selected steel is center marked on both ends and is of a length calculated to produce a given footage

of pipe. It is placed in a soaking furnace, where it is slowly brought up to a uniform temperature of proper value for hot working.

From the furnace, the "tube round" is positioned on the piercing unit of the hot mill, where it is forced by a set of spinning rolls over a piercing mandrel, as illustrated in Fig. 7, thus producing a rough hollow tube with a relatively even wall, of somewhat greater length than the original tube.

While still hot, the pipe moves on to a sizing roll that kneads it over a mandrel bar. The bar and the pipe then pass through a series of rolls that work it to the desired outside dimension. After this, the mandrel bar is stripped from the inside, and cooling, straightening, and inspection of the pipe follow.

The newer mills, now under construction, will have, in addition to the steps enumerated, a subsequent step called "stretch rolling." A stretch rolling mill will elongate and reduce. Thus it will be possible for the hot mill, for example, to deliver a 2-inch pipe to the stretch mill, which will automatically produce a 1½-inch, 1¼-inch or, in some cases, a 1-inch pipe at greater speeds than can be obtained by other methods.

CARGO HANDLING WITH OVERHEAD CRANE GEAR.

The following is a partial reprint of a technical paper by A. C. Rohn, I. W. Smith, and G. B. Hanes of the U. S. Maritime Commission which was published by the Alumni Association of the U. S. Merchant Marine Cadet Corps.

In analyzing overhead cargo gear and attempting to improve its efficiency, it is apparent that

1. (a) More loads should be moved per unit of time.
   or
   (b) Heavier lifts should be handled.
2. Loads should be more easily picked up from any position on the dock and more easily spotted in the hold.

FIGURE 1.—The *Sea Hawk*, a C-3 Operated Experimentally by the U. S. Maritime Commission.

FIGURE 2.—Photograph of Model Used in Developing the Gear.

Inasmuch as hook speeds of the conventional boom gear have about reached the maximum as limited by the reflexes of the winch operator, the most fertile points for consideration are 1(b) and 2 above.

The overhead crane gear is a development to meet these requirements. This gear, which is essentially a modified shop crane placed on runways above deck, has been installed over the three forward hatches of the *Sea Hawk*, a C3 vessel, shown in Figs. 1, 5 and 6, built by the Ingalls Shipbuilding Corporation at Pascagoula, Miss. A photograph of a model, (Fig. 2), used in developing the gear shows more clearly the working parts and will be used in describing its operation.

The crane, carrying two trolleys, runs athwartship on runway girders and out over the ship's sides on cantelever trusses. These trusses are hinged so they can be swung inboard and stowed when at sea. Fore and aft movement of the hooks is accomplished by movement of the trolleys on the crane bridge girder. Machinery for the hoist and for trolley travel are contained in the trolley. All controls are centered on the bridge girder, thereby allowing the operator a clear view of the hooks at all times.

One five-ton trolley is installed over hatch No. 1 and two five-ton trolleys over hatch No. 2. Each of the two trolleys over No. 3 hatch is equipped with a three-ton and a fifteen-ton hook. Hook selection is made simply by throwing a single lever on the trolley. This arrangement allows fast lifts of loads up to three tons or slower lifts of up to fifteen tons; by using an equalizing strong back a 30-ton lift can be accomplished by two hooks without any change in rigging. Considering that several hours is not an excessive time requirement for the present rigging of the conventional jumbo boom, this feature is considered quite important in cutting down port time.

The conventional boom gear has been retained on the after holds of the *Sea Hawk* in order that a direct comparison of this gear with the overhead crane gear working under identical conditions could be made.

As soon as the boom gear and the overhead gear above No. 3 hatch were installed, extensive cycle tests were made to obtain a comparison of the efficiencies of the two gears. The results of the test are shown in Fig. 3.

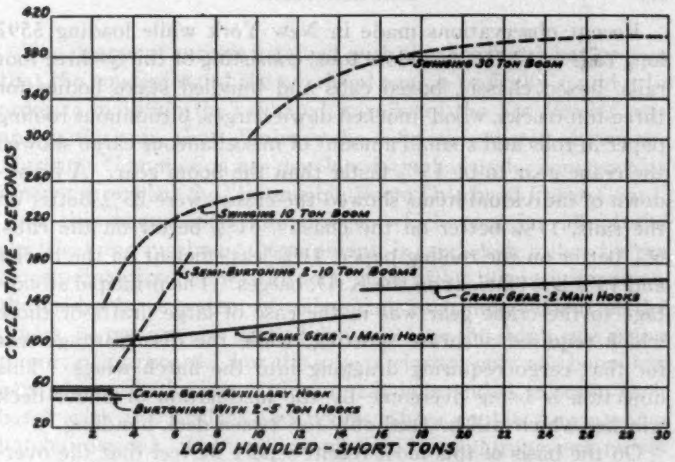


Fig. 3

Cycle times shown include hooking and unhooking loads, but do not include handling and stowing in the hold.

Briefly, the tests showed that burtoning loads up to 3 tons with the boom gear is about 10% faster than handling these lighter loads with the crane gear. For loads above 3 tons, where it is necessary to employ semi-burtoning or swinging of the conventional gear, the crane gear shows an increasing advantage as the loads are increased. Also the tests clearly demonstrated the greater ease with which the crane gear could spot the loads at any position in the hatch or on the dock.

During the first several months of actual service many minor faults developed in the crane gear. A number of these faults were due to difficulties peculiar to shipboard operation which were not foreseen in the design. The unfamiliarity of the ship's crew and longshoremen with the new gear and its intended operation has accounted for the development of many minor mechanical troubles. The early operating reports were, therefore, discouraging to those who had hoped that this trial installation would function perfectly and be a cureall for cargo handling problems. Due to the splendid cooperation of the crane manufacturer, the operating company and the individual members of the ship's crew, it has been possible to remedy the majority of the mechanical difficulties.

Recent observations made in New York while loading 5597 long tons or 12,770 stevedore tons, consisting of thirty-three foot rails, boxed chassis, boxed cabs and bundled stake bodies for three-ton trucks, wood knocked down barges, bituminous roofing paper in rolls and a small amount of miscellaneous cargo showed the crane gear to be 15% faster than the boom gear. A breakdown of individual items showed the cranes were 23% better on the rails, 17% better on the chassis, 31% better on the cabs, 6% better on the roofing paper, 11% less efficient on the bodies and 13% less efficient on the K. D. barges. The principal advantage to the crane gear was in the case of large drafts or those which required accurate spotting, while the disadvantage was for that cargo requiring dragging into the hatch wings. This objection is being overcome by the installation of under deck winches which will be used only for 'tween deck handling.

On the basis of this most recent report we feel that the overhead crane gear is reaching that stage of development where it can be called a success. Again let us repeat that this success has been made possible not by any individual or any group, but by all individuals from the designer to the crane operator cooperating and doing their best to work with a new idea, a new gear.

The above report paints quite a rosy picture in showing that the gear has fulfilled its intended purpose. However, as is true in almost any radical design, it has certain basic faults or limitations which are briefly as follows:

1. The overhang of the cantilever jibs is fixed so that very wide camels are required at docks having narrow aprons. These camels are not particularly objectionable except when they are used in slips intended for twin berths.
2. The fixed height of the crane limits the drift. In the case of loads with long slings the drift provided is inadequate.
3. Hatches Nos. 2 and 3 are equipped with two trolleys to allow double ganging of these hatches. Both of these trolleys being carried on the same bridge results in the transverse movement of both trolleys being governed by the speed of the slower hatch gang.
4. The transverse movement, which is a great per cent of the total movement, is relatively slow due to the great mass of the load, the bridge and both trolleys.

The degree of success attained with the *Sea Hawk* has proven that the conception of the overhead gear is basically sound. In order to maintain the principal features of the *Sea Hawk's* gear and at the same time eliminate the deficiencies listed above, the Maritime Commission has developed and has under construction a new gear called the "Extensible Boom Overhead Cargo Gear."

In this new design, the bridge, which contains all of the electrical and mechanical equipment, is carried on rollers by the longitudinal fixed runway girders. Extensible booms hung under the bridge by a system of rollers can, each independent of the others, be put out any desired distance over the ship's side either to port or starboard. For the sake of clarity, only one boom has been shown in the illustration, but it is intended to equip each hatch with two bridges and booms where double ganging of a hatch is desired. The load is supported by, and transverse movement attained by, movement of a light flying trolley which runs in the boom. See Fig. 4 showing cut away section of boom and trolley.

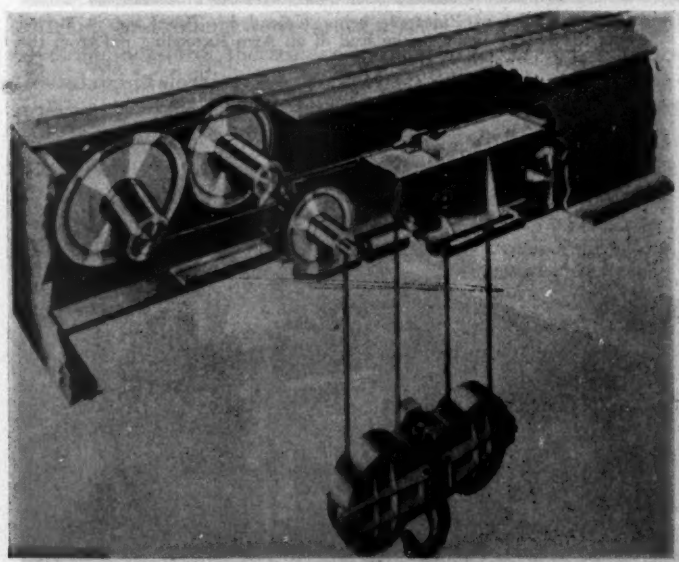


FIGURE 4.—Cut-away Section of Boom and Trolley.

FIGURE 5.—A Pallet of Pineapple being Unloaded. The Load is Moved Along the Heavy Beam to the End of an Arm Extending out from the Ship and is Lowered to the Dock.

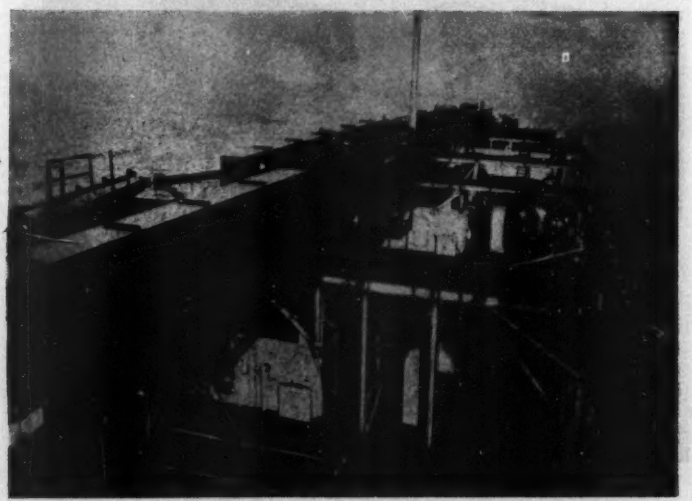


FIGURE 6.—C-3 *Sea Hawk* with Cargo Gear Secured for Sea.

Another feature illustrated is the use of twin hatches, which practically eliminates hatch wings and allows any load to be spotted directly in its final position of stowage, thereby eliminating handling in the 'tween decks.

The gear now under construction is designed to carry about seven tons on a two-part hook block or fifteen tons on a four-part block, thus making it possible to handle a single 30-ton load on two bridges over one hatch. Changing from two-part to four-part hoist requires no rereaving, but is accomplished by merely placing the line over an idler pulley on the flying trolley.

Practical considerations, such as wheel house vision and stability, limit the height of the fixed runways and the drift. In this new gear about two feet more drift is obtainable than on the *Sea Hawk's* crane gear. Whether or not this is satisfactory can only be determined by actual fair and unprejudiced trials in service.

In order that as many of the minor deficiencies be corrected as possible before actual installation aboard ship, a land test structure simulating as nearly as possible an actual ship is being built. This structure can be tilted so as to simulate normal list. Pitching and rolling, however, cannot be simulated, nor can the dynamic action of the seas or the inertia forces sometimes encountered in docking.

We say advisedly that many of the minor deficiencies will be corrected. About all that the land installation can prove is whether or not the general conception of the gear is practicable. In so radical a design, it is impossible to foresee all the problems that will arise under service conditions. These problems must be recognized and solved, if possible, as they arise.

The two gears described above are undoubtedly not the final answer to our cargo handling problems. Time may prove that they in themselves are of no importance in the final solution of our problems. What is important is that a real attempt is being made to increase the efficiency of our ships; and by all of us—you men who run the ships, the operators, we designers and all others—keeping an open mind and cooperating we cannot help but attain our goal of the Best Merchant Marine in the World with Security For All of Its Workers.

FLAME HARDENING — PRINCIPLES, APPLICATIONS, AND EQUIPMENT.

The following article by M. S. Rosengren is reprinted from "The Welding Journal" for June, 1948.

INTRODUCTION.

Probably no one knows exactly when flame hardening was first used. Fundamentally, its accomplishments today are no different from the results obtained by Roman soldiers in tempering their war weapons. These were simply heated in a fire until they were of a bright red color, and then quenched by being plunged into the bodies of soldier's slaves. Although socially cruel, this method was metallurgically adequate, and involved no chemical change with regard to the steel—merely a reorientation of its physical characteristics. To this day, flame hardening adheres to the same principles.

It remained for the rapidly expanding use of acetylene in combination with oxygen to establish flame hardening as a specialized method for heat treating hardenable ferrous alloys. The natural wedding of the terms "flame hardening" and "oxyacetylene" took place because of two inherent characteristics of the combined gases—the high temperature obtainable, and the ease and flexibility of application to a wide variety of shapes and sizes of work.

Nature has decreed that iron, when alloyed with other elements —particularly carbon—exhibits certain physical characteristics. During the process of heating a piece of this alloy, these characteristics will change according to definite laws at definite points on the temperature scale. During the process of cooling, the same piece of alloy will, in general, show these same characteristics in the reverse order at other definite points on the temperature scale. Just to complicate the picture, there are other tendencies of a physical nature inherent in this alloy during the cooling process, providing that the temperature from which the cooling process starts has exceeded a certain point, usually referred to as the upper critical limit. These other tendencies may be thought of as stages through which the steel must pass in order to revert again to its initial condition. Years of experimentation have shown that the time during which

cooling takes place is a very important factor, and that by varying this time, certain of these tendencies can be arrested so that when the steel has attained a normal room temperature at which it can be handled and utilized, it will still have these tendencies as definite physical properties. One of these is extreme hardness, a physical characteristic not exhibited by steel in what is referred to as its normal state, but definitely attainable by arresting it at a certain stage during the cooling process. This must be done by accelerating the cooling rate by use of a quenching medium that will cool the piece more quickly through the critical range than would the ordinary surrounding medium of air at normal temperature.

The many processes of heat treatment of steel for hardness including flame hardening, utilize this principle to accomplish the desired result. The ordinary heat-quench procedure generally accomplishes this change throughout, or nearly throughout, the entire piece of work. Unfortunately, the harder a piece becomes when so treated, the more brittle it becomes also, as these two tendencies occur simultaneously. Thus when a piece is hardened, it is also embrittled for the extent of the hardened zone, thus lessening its ability to withstand shock and impact. Further treatment, called drawing or tempering, is usually required to lessen this tendency, which is accomplished only at a sacrifice of surface hardness.

This objectionable situation is alleviated by flame hardening. It is possible with flame hardening to obtain the maximum hardness which the steel is capable of producing, in the treated portion only without disturbing the balance of the metal. The most common reason for flame hardening is to effect a wearing surface with high abrasion-resistant qualities in combination with a tough, ductile and shock resistant core. In short, flame hardening can be applied only where needed.

This leaves the remainder of the work in the same state after flame hardening as it was before, with all of its original properties intact. If through design demands, it is necessary to give a machine part, for example, a preliminary heat treatment for considerations other than hardness, the imparted characteristics will in no way be disturbed by flame hardening afterward, except in the area where the hardness is desired.

The hardened case itself does not stop abruptly in depth. There exists between it and the unaffected portion a buffer or transition zone in which the physical characteristics graduate from those of the hardened case to those of the metal in its pre-flame hardened condition. This serves to bond the case to the piece and lessen the tendency to spall by being capable of absorbing the stresses induced by quick quenching.

Flame hardening is generally applied to the metal by means of a suitable burner arrangement followed by any of various means of quenching medium application, such as a hose for water, a block with a drilled face for air, water, brine or a soluble oil mixture, or a tank in which the work can be immersed in any of several quenching media. Usually some type of mechanism is involved to effect movement of the flame and quench along the work, or the movement of the work past a stationary flame and quench arrangement. It is customary to divide these types of application into four methods—Stationary, Spinning, Progressive and Combination.

The Stationary Method.

Sometimes referred to as spot hardening, the stationary method involves the heating and quenching of a small area which can be treated without the necessity of differential movement between the work and the flame-quench apparatus. Examples are the hardening of the ends of valve stems, or spots on bearing buttons. Quenching can be accomplished by flooding the piece after removal of the flames, or dropping the work into a quench tank. This is very often accomplished by hand held equipment.

The Spinning Method (Fig. 1).

The spinning method is used when hardening relatively small parts which lend themselves to rotation, such as small shafts, pulleys, rollers and gears, pinions or sprockets with small pitch teeth. This is done by mounting the work on a spindle and revolving it at a constant Rpm. Heating heads are mounted around the work's circumference and the piece is heated for an experimentally determined length of time, after which the piece is quenched by any of several methods. The quenching appara-

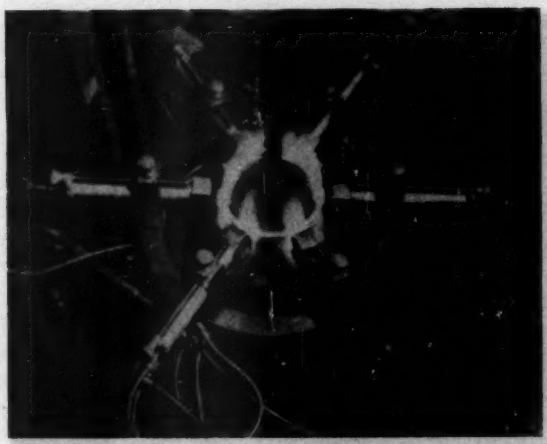


FIGURE 1.

tus may be so arranged that it can displace the heating heads after the fire is turned off, and play the quenching medium directly on the rotating work or the work may be ejected from the spindle into a quenching tank.

THE PROGRESSIVE METHOD.

The progressive method refers to the progressive movement of flames and quench along the work which is held stationary—or the reverse of this, the progressive movement of the work past the stationary flame and quench apparatus. The former would include such work as the flame hardening of machine tool ways, punches, dies and gear teeth (Fig. 2.) of relatively large diametral pitch as well as sprockets with large teeth. The latter includes traction and crane wheels, (Fig. 3) pulleys, sheaves, worms (Fig. 4) and cams, where the work is rotated slowly past the flames and quench, the full hardening operation being accomplished with one revolution of the piece.

THE COMBINATION METHOD.

As its name implies, the combination method involves the movements of both of the two previous methods and is used for the hardening of long cylindrical shapes such as shafts, rolls

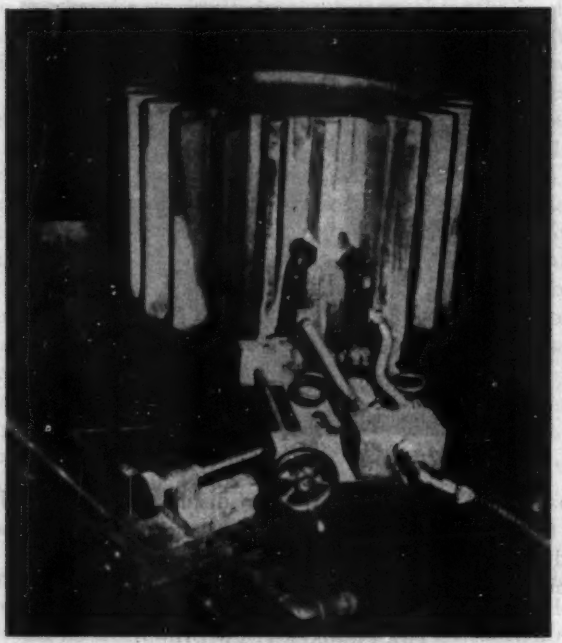


FIGURE 2.

and boring bars (Figs. 5 & 6). The work is mounted between centers or in a chuck, revolved at a constant Rpm. and the flames, followed closely by the quench, move along the longitudinal axis of the work at a rate of speed consistent with the desired result. The method is also widely used to harden in inside cylindrical surface such as gear and pulley bores or pump and cylinder liners.

APPLICATION.

Any of the above outlined methods may be used successfully on a wide variety of ferrous materials—steels, cast iron and Meehanite. The most essential requirement is the carbon content, which should be no less than 35 points. This should be construed as combined carbon for the cast metals. Best results are obtained when the carbon content is between 35 and 55 points. A greater carbon content than 55 points should be

FIGURE 3.

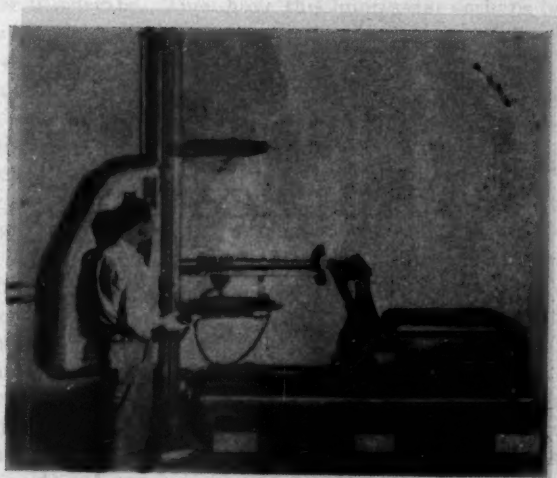


FIGURE 4.

FIGURE 5.

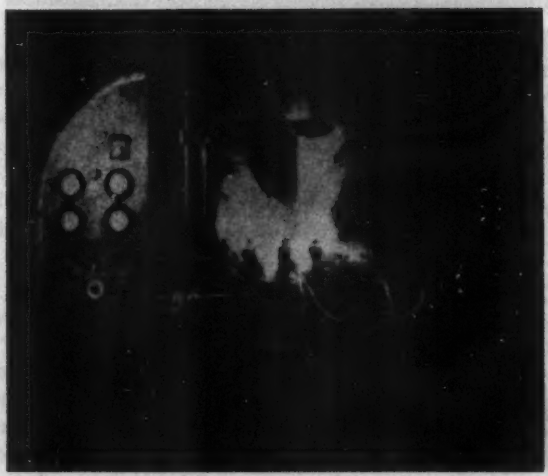


FIGURE 6.

treated with care, as there is danger of cracking from overheating or too drastic quenching. The addition of other alloying elements in moderate amounts has no appreciable effect on the end result, but aids the process by widening the critical ranges of both heating and cooling.

Manual operation of heat and quench, using a hand torch and a water hose or a quenching tank, has been and is being used for flame hardening. However, the present tendency is toward automatic application of the process in order to duplicate results.

Flame hardening has been looked upon in the past as something of a necessary evil. It has been tried and found wanting in many instances and subsequently dropped. Much of the trouble encountered in such cases was simply lack of control. In other cases, efforts have been made to better the product by bettering the process of application. Ten years ago, R. F. Arnoldy, now operating the Houston Heat Treating Co. in Houston, Texas, had this to say of the process:

"Throughout industry, to a large degree, however, the process still has a 'blacksmith shop' status. To some shop superintendents flame hardening is something you do with a collection of pipes and gadgets which takes up a lot of room in his plant, ties up his machines, costs a lot of money and takes a Philadelphia lawyer to operate. They have this impression perhaps because flame hardening equipment has not completely advanced to the precision machine stage as yet, and when mounted on a machine tool does not give a very production- or precision-like appearance."

This same thing could be said about it in many quarters today. However, the late war gave flame hardening a stimulus that no other single factor could have. Many people used it successfully through that period who would normally have nothing to do with it. Its tremendous potentialities were finally recognized, and by careful thought and planning, it has made great strides in the last five years. Much thought was given and work done on the problem of adequate control and the standardization of the multiplicity of variables involved in its use. Today, under the careful guidance of the engineer, it is rapidly attaining maturity alongside other and older methods of heat treating, and is becoming recognized as a standard specification at the drawing board.

The development of this control was not easy. It has been recognized for sometime that the flow of the gases should be proportioned accurately both from the standpoint of the ratio of one to the other in the mixture, and with regard to the total number of cubic feet consumed in any given unit of time, in order to assure duplication of results. Valving arrangements involving fixed or variable orifices to control the flow of oxygen have been and are being used. Development is going forward daily on an instrument that will hold a constant proportion of the gases at all times. Both mechanical and hydraulic operation has been used to move the flame and quench apparatus with reference to the work, as well as some adaptations of welding and cutting equipment. Spindle speeds are varied by the innovation of the thyratron controlled d.-c. motor giving infinite variation of Rpm. over a great range. The most significant single development has been the use of a thermopile to measure the intensity of radiation given off by the piece during the heating cycle. By using the variable output of current from the thermopile through an electronic circuit, usually in a controlling-recording instrument, many subsequent mechanisms can be activated through a series of relays to assure the operator that the piece is not being over or under heated, and that it is quenched at just the proper time to obtain the end result.

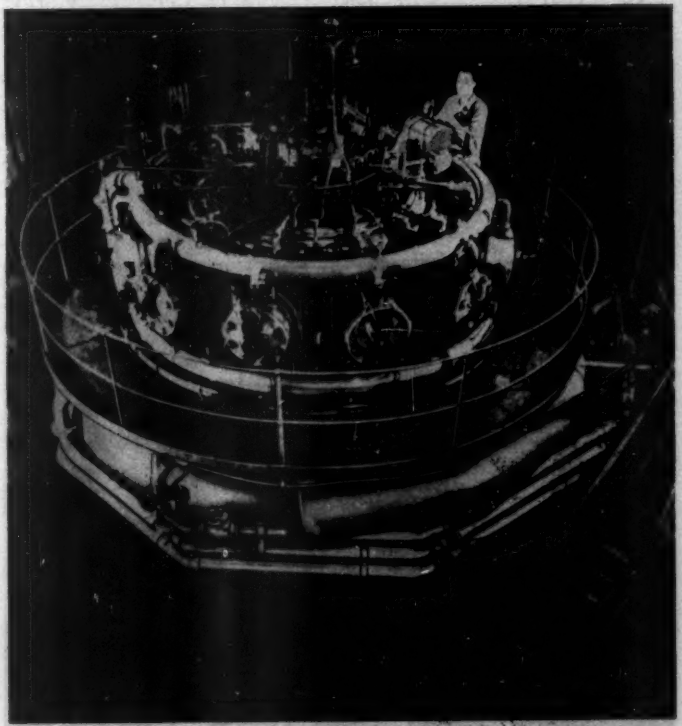
THE NORDBERG RADIAL ENGINE.

The following article is reprinted from "Motorship" for July, 1948.

A new engine, with cylinders arranged radially about a vertical crankshaft, has been manufactured and proven by the Nordberg Manufacturing Company.

The success achieved by the pilot model in stationary electric generating service would seem to recommend this engine for marine application on diesel-electric propelled motorships.

The Nordberg radial engine is built as a spark fired gas engine with a patented system of spark ignition, as a diesel engine and as a dual fuel engine. Its patented features of design are not found in conventional diesel engines and they offer certain advantages that merit the serious attention of marine engineers.

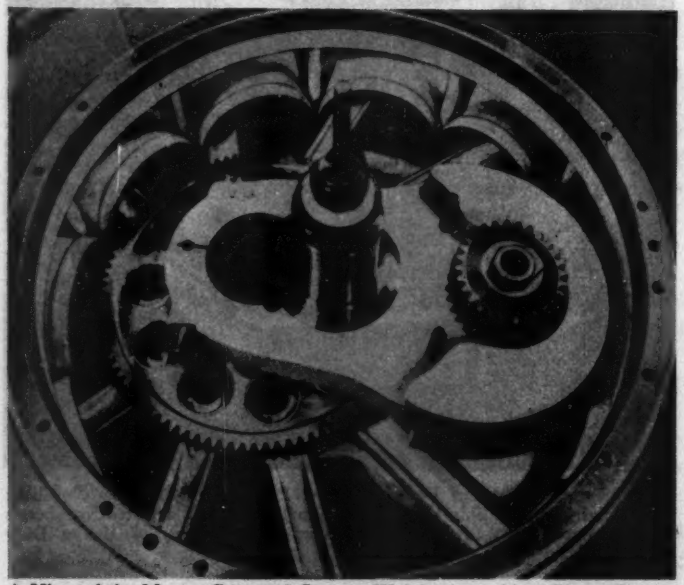


The Operator at Control Position on the Platform of the New Nordberg Radial Engine. This View of the Complete Unit Illustrates Nordberg's Striking Departure From the Conventional Design of Heavy Duty Engines.

Built with eleven cylinders of 14-in. bore and 16-in. stroke, this engine develops 1800 Hp. or 1250 Kw. at 400 Rpm. The scavenging blower motor requiring approximately 100 Kw. input is connected directly to the leads of the generator making a self

contained unit automatic in its operation, leaving a net capacity of 1150 Kw.

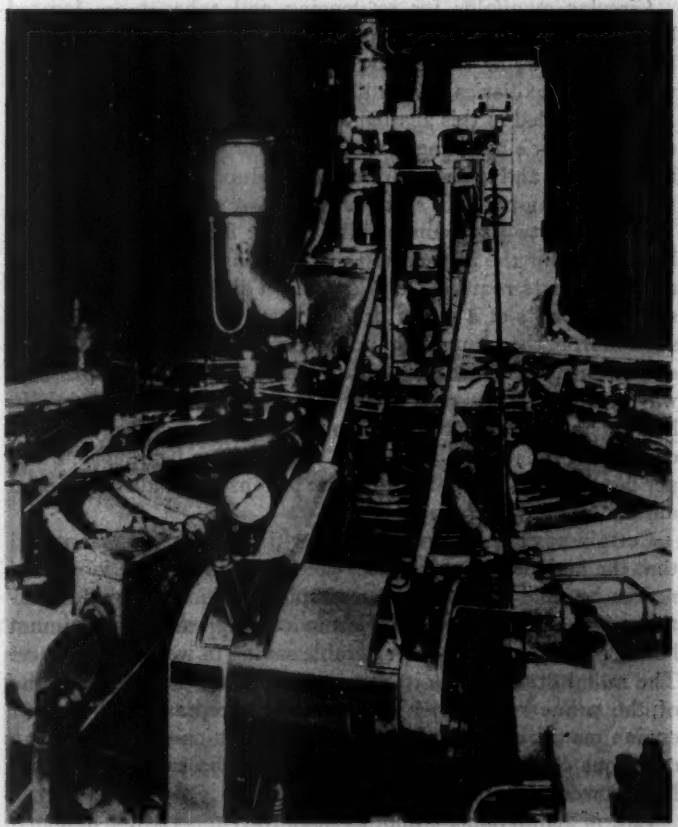
The compact design of this engine provides for direct connection to the generator which is mounted at the lower end of the vertical crankshaft in the coolest zone. This arrangement requires headroom of less than twenty feet measuring from engine foundation and little more than twelve feet laterally. On board ship where a saving in space is generally reckoned a saving in money, the relatively small space requirements of this 1800 Hp. unit recommend it for motorships using diesel-electric drive. A further consideration in the matter of space-saving is the fact that cylinder parts can be serviced from the floor eliminating the necessity for clearance above the engine to pull pistons and doing away with the usual heavy crane and supports.



A View of the Master Gear and Counter-Weight Assembly. Eleven Bronze Bushed Knuckle Pins in the Master Gear Take the Thrust From the Connecting Rods.

The Nordberg radial engine operates on the simple two-cycle principle with port scavenging and port exhaust. Power is

transmitted through connecting rods to a master gear mounted on the single crankpin of the shaft. The master gear assembly with its eleven bronze bushed knuckle pins is kept constantly aligned through pinion gears meshing with the master gear and a stationary gear attached to the heavy cover bolted to the top of the frame. The relationship of the various parts that is maintained by the gearing assures the optimum position of the master



A View From the Control Position Showing the Dual System of Controls for Operation on Either Gas or Oil.

gear and counterweights for the absorption of piston thrust and balance of forces.

The only other main bearing required is the lower one located in a hub which is part of the frame. A thrust bearing to carry the weight of the crankshaft is also incorporated. The upper main bearing is in the heavy symmetrical cover which is bolted to the cast main frame. Mounted on this cover are the governor, fuel pumps, mechanisms for oil and gas operation and controls.

Circular manifolds for scavenging and exhaust are located below the operating platform. The motor driven centrifugal blower discharging scavenging air into the manifold is included with other auxiliaries grouped in a unit at the engine base.

Also included in this auxiliary group is a motor driven centrifugal pump for circulating cooling water through the cylinder and head jackets and a motor driven rotary lubricating oil pump for the circulating pressure system.

There are two systems of lubrication: force feed mechanical lubricators for the pistons and a circulating pressure system for all other parts requiring lubrication. This circulating pressure system consists of a motor driven rotary pump which takes its suction from the sump tank and discharges into the lower bearing of the shaft. From this bearing, the oil enters the drilled passageway in the shaft and rises where it is distributed to the crankpin bearings in the master gear, through drilled passageways in the connecting rods to the wrist pins, to the pinion bearings and upper bearing for the crankshaft. The pistons are made of a special aluminum alloy and are oil cooled.

Certain advantages are claimed for this design. It is reported that the high degree of balance, which tends to reduce vibration, makes possible mounting on a simple and inexpensive foundation. As mentioned above, the compact design requires minimum headroom and achieves considerable saving in weight and space. The radial design lends itself to weight reduction which is one of the prime reasons for its universal acceptance by aircraft engine manufacturers. It is notable in this case that, with no conscious attempt at weight reduction, the unit weight per horsepower is approximately 47 lb.

Ease of maintenance is another advantage reported. The short, counter-weighted crankshaft requires maintenance on only two bearings, both of which are renewable bushings easily replaced

at minimum expense. None of the bearings on the engine are of the split type. Easy maintenance is further facilitated by making all heads, pistons and cylinders interchangeable, a feature which also reduces the number of spares to be carried to a minimum.

As noted, the engine is designed to operate on gas or oil, whichever is available or more advantageously priced. At present, it is reported that some 80 units are on order for the Texas plant of the Aluminum Company of America. These will be used to supply this operation with an output of 70,000 Kw. and it is intended to use natural gas as fuel.

The Nordberg radial engine appears to represent a considerable advance over existing practical designs. Although its application thus far has been confined to stationary land installations, its manifest advantages commend it for application in diesel-electric powered motorships in the early future.

A METHOD FOR THE DETERMINATION OF A SHIP'S STABILITY AT SEA.

The following paper by Capt. J. E. Kiernan, U. S. N. (Ret.) was presented at the Spring Meeting of the Society of Naval Architects and Marine Engineers.

These notes give the results of experiments made with a 20-foot model of a large vessel. The object of the tests was to determine the stability of the model from measurements of its rolling motion in irregular waves, without knowledge of loading conditions.

Fig. 3 gives the results of the experiments (Runs 154, 156, and 162). In each case the two dotted lines represent stability curves predicated upon rolling data obtained by two instruments described later. The full line in each case is the stability curve based on the location of the center of gravity as determined by an inclining experiment.

The method of determination of the stability curves is based

on the fact that, if the cross curves of stability are known, the stability curve can be found, if the displacement and location of center of gravity are known. In Appendix 1 it is shown how the center of gravity can be located without knowledge of loading details, if the angle of heel of a ship (if any) and the slope of the stability curve at that angle are known. This slope can be found from the natural period of roll of the ship by equation (4), Appendix 1. Thus, the only information required to locate the center of gravity is the knowledge of the ship's displacement, period of roll, and angle of heel, if any.

It is assumed that the ship's displacement can be found with fair accuracy from draft gages and that the angle of heel (average list, if the ship is rolling) can be determined by available or procurable means. However, for these experiments, displacement and list were determined in still water and only the natural period of roll was determined by measurements in a seaway. The determination of the natural period from the actual roll of the ship is the most difficult step in the procedure. At present two methods are being used:

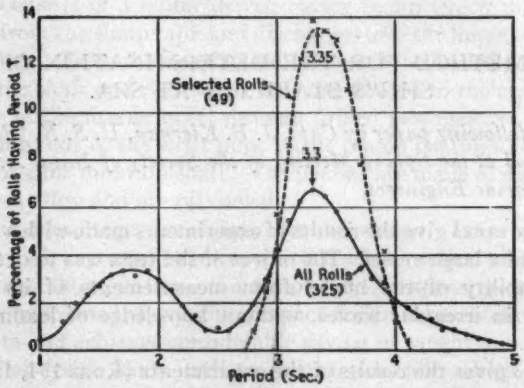


FIG. 2.—DISTRIBUTION CURVES TO DETERMINE NATURAL PERIOD.

(*1*) *Roll Recorder.* The angle of roll of the ship is recorded and plotted versus time. From this record periods of successive rolls can be found. If the latter are tabulated and distribution curves

plotted (percentage of rolls having a given period plotted against that period), the most often occurring period, the peak of the distribution curve, is invariably close to the natural period. If a selection is made, tabulating only rolls of fairly good wave shape and not too large an amplitude (the latter being forced rolls in all likelihood), sharper distribution curves result.

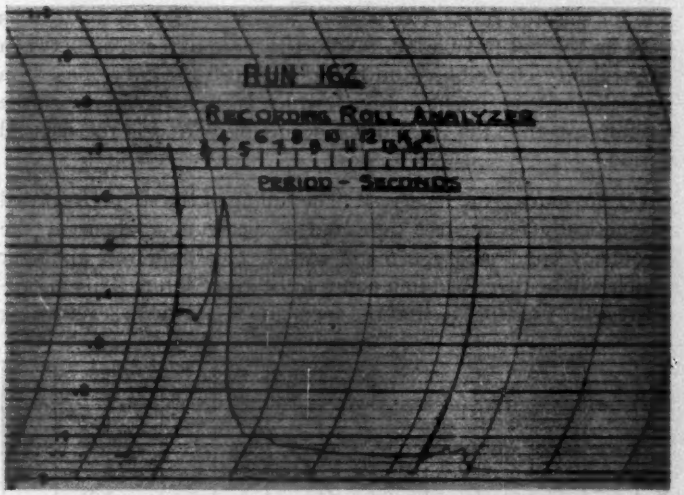


FIGURE 1(a).—Sample Recorder Chart.

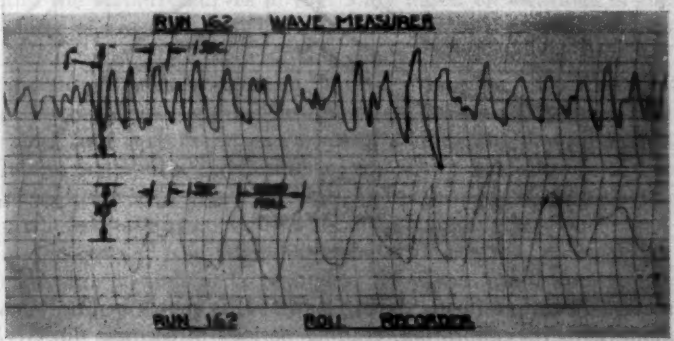


FIGURE 1(b) and (c).—Sample Recorder Charts.
(b) Upper Curve—Wave Measurer. (c) Lower Curve—Roll Recorder.

For example, Fig. 1(c) represents a part of an actual recording obtained during Run 162. A roll better than average is indicated. Fig. 1(b) is a recording made simultaneously of the wave action at a point near the model and thus gives some idea of the waves which the model encountered.

Fig. 2 shows the distribution curves plotted for this same experiment (Run 162), one curve being drawn for all rolls and one for selected rolls. The peak of the latter (3.35 seconds) was used as the natural period. Note the close agreement between the stability curve based on this period and the curve based on an inclining experiment in Fig. 3, which is in conformity with the assumption that the most often occurring period is the natural period.

(*2*) *Recording Roll Analyzer.* The rolling of the ship is recorded magnetically on a steel tape and a spectrum analysis of the frequency components is made with a recording roll analyzer which will be described in a paper to be given later. This instrument is fully automatic and the spectrum is drawn by a recording

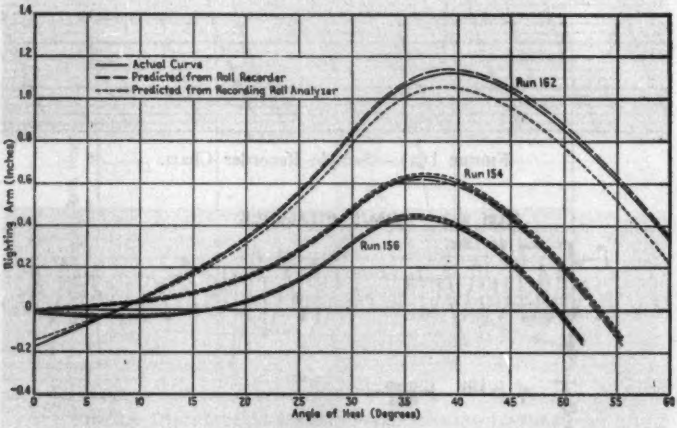


FIG. 3.—STABILITY CURVES

meter. A sample (made during Run 162) is shown in Fig. 1(a), along with its calibration, reading periods in seconds directly. Because the tendency to roll at the natural period is always

present, some excitation at the natural frequency will always appear on this record and tend to give it a measurable peak.

Once the slope of the stability curve, angle of heel, and displacement are determined, the location of the center of gravity can be found from equations (8) and (9) given in Appendix 1 and the stability curve plotted from equation (1).

TABLE 1

Run No.		154	156	162
Displacement, lb.		1220	1220	1220
Natural period from roll recorder, sec.		9.3	6.0	3.35
Natural period from recording roll analyzer, sec.		8.3	6.2	3.6
Angle of heel, deg, ϕ.		0	15.8	7.8
GM, in.	Determined from inclining experiment.	0.215	−0.094	1.225
	Computed from roll recorder.	0.193	−0.067	1.245
	Computed from recording roll analyzer.	0.237	−0.091	1.086
Transverse position of center of gravity, in.	Determined from inclining experiment.	0	0.016	0.172
	Computed from roll recorder.	0	0.016	0.168
	Computed from recording roll analyzer.	0	0.009	0.146

The results of the analysis are shown in tabular from in Table 1, and the stability curves in Fig. 3 are based on locations of the center of gravity as given in Table 1. Note that Fig. 3 shows greatest accuracy in determining characteristics for conditions of low stability where accuracy is needed most.

The methods described in this report assume that the hull is intact and that the cross curves of stability apply. Dynamic measurements then give information which helps to locate the virtual center of gravity. Thus, if free surface is present, the predicted stability curve does not include the effects of pocketing; in the case of free communication with the sea, considerable errors are possible at large angles. However, a fair approximation of the stability curve at small angles seems always possible.

If the ship trims considerably, enough to affect the stability, then the amount of trim must be known and data must be avail-

able (either computed from the shape of the hull or measured from model experiments) to duplicate the cross curves of stability at all possible values of trim. Enough static experiments involving trim have been made by this group to prove their usefulness and the desirability to complete such experiments.

Note that in computing the slope of the stability curve from equation (4) in Appendix 1 the radius of gyration k had to be known. For these experiments it was taken as the value which existed during still-water rolling at the same displacement and period of roll. A further analysis of the large amount of data available is to be made to determine the variation in k and to develop methods for the prediction of its value for use in future work.

APPENDIX 1.

THEORY.

For the known displacement of the ship, let curve A (Fig. 4) represent the stability curve as determined from the cross curves of stability. Applying a sine correction, curve B may be deter-

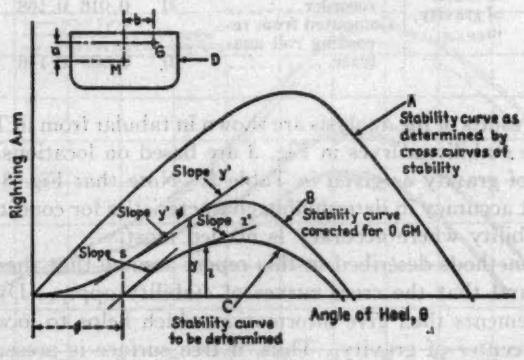


FIG. 4

mined for assumed zero GM (center of gravity at M). Let y be the ordinate of this curve (righting arm) and let

$$y' = \frac{dy}{d\theta}$$

be the slope of this curve. Let curve C be the actual stability curve of the ship, the curve to be determined. Let its ordinate (righting arm) be z and its slope

$$z' = \frac{dz}{d\theta}$$

Assuming the hull to be intact, the new curve C differs from B only in so far as the actual center of gravity is not at M. In other words, the ordinate z may be found from y by applying a sine and a cosine correction. Assuming the center of gravity to be at a point G (see Diagram D, Fig. 4),

$$z = y - a \sin \theta - b \cos \theta \tag{1}$$

and

$$z' = y' - a \cos \theta + b \sin \theta \tag{2}$$

It is known that the list of the ship is ϕ, hence when

$$\left.\begin{aligned} \theta &= \phi \\ z &= 0 \end{aligned}\right] \tag{3}$$

Also, it has been found from model experiments by this project that the slope of the stability curve can be determined from the natural period of roll through small angles by the formula

$$s = \frac{(2\pi k)^2}{gT^2} \tag{4}$$

where

s = slope of stability curve at equilibrium angle, inches per radian. (This is equal to GM when the angle of list is zero.)

k = radius of gyration, inches.

T = period, seconds.

g = acceleration of gravity, inch per sec^2.

Hence, slope s at angle ϕ can be determined and for

$$\left.\begin{aligned} \theta &= \phi \\ z' &= s \end{aligned}\right] \tag{5}$$

substitute conditions (3) and (5) in equations (1) and (2):

$$0 = y\phi - a \sin \phi - b \cos \phi \tag{6}$$

$$s = y\phi' - a \cos \phi + b \sin \phi \tag{7}$$

Solving equations (6) and (7) simultaneously yields

$$a = (y\phi \sin \phi + y\phi' \cos \phi) - s \cos \phi \tag{8}$$

$$b = (y\phi \cos \phi - y\phi' \sin \phi) + s \sin \phi \tag{9}$$

Once a and b are determined from equations (8) and (9) the stability curve may be plotted easily from equation (1).

Note that in equations (8) and (9) the quantities in parentheses could be computed and tabulated or plotted for all displacements and angles. Then, in solving a problem, solutions for a and b, the coordinates of the center of gravity, could be greatly expedited.

APPENDIX 2.

In the work on the determination of the stability of ships the theory has been advanced that in an irregular sea the most often occurring period of roll would be the natural period. This might be accepted, if it is agreed that the term "most often" does not necessarily mean a majority of the rolls but only more than any other one period. Also it must be admitted that in a regular sea a ship eventually will be forced to roll in or close to a period corresponding to that of the waves. However, the information is ever present so that any one roll is a combination of the natural period and other influences and thus the problem becomes one of isolating what is the natural period.

This feature of the natural period was observed during experiments made on the *Physalia*, a 60-foot V-bottom motor sailer, and the *Asterius*, a 40-foot V-bottom fishing boat, through the courtesy of the Woods Hole Oceanographic Institute, from July 11 to July 19, 1946. In this case the natural period of roll was checked by sallying the ships in still water.

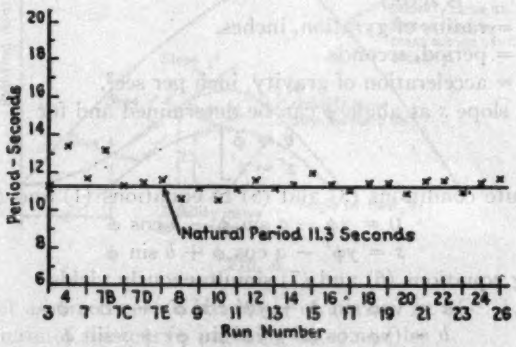


FIG. 5.—RECORDING ROLL ANALYZER, PERIOD OF ROLL, S. S. "CELIMENE"

A further investigation of this was made on the French tanker *Celimene*, through the courtesy of the Bethlehem Shipyard, Hoboken, N. J. The ship is a Maritime Commission T-2 tanker and on this trial had drafts of 15 feet 5 inches forward, 18 feet 2 inches amidships, and 21 feet 3 inches aft. The trials were held in the lower bay of New York and the results of the analysis are shown in Fig. 5. These bear out the fact that the most often occurring period is at or near the natural period of 11.3 seconds. This natural period was determined by causing the ship to roll freely in the smooth water of the river by means of a tug before proceeding with the trial.

REAR ADMIRAL CLAUD ASHTON JONES, U.S.N.

OBITUARY.

Rear Admiral Claud Ashton Jones, U. S. N. (Ret.).

Rear Admiral Claud Ashton Jones, U. S. Navy, Retired, member and past president of the Society, died of a cerebral hemorrhage at his home in Charleston, West Virginia, on August 8th, 1948. Admiral Jones, holder of the Congressional Medal of Honor and the Legion of Merit awards, was 64 years old.

Admiral Jones was born in Fire Creek, Fayette County, West Virginia, on October 7th, 1884. After attending schools in Fayetteville and Charleston, West Virginia, he was appointed to the U. S. Naval Academy in 1903. He graduated with the class of 1907 in September, 1906 and made the cruise around the world in 1907 on the U. S. S. *New Jersey*. Following post graduate instruction in electrical engineering he received from Harvard University the degree of Master of Science in 1913.

In 1916 he was aboard the U. S. S. *Memphis* on August 29th when that vessel was wrecked by a tidal wave in the roadstead at Santo Domingo. For his heroism during this disaster he was later awarded the Congressional Medal of Honor with the following citation:

"For extraordinary heroism in the line of his profession as Senior Engineer Officer on board the U. S. S. *Memphis* at a time when that vessel was suffering total destruction from a hurricane while anchored off Santo Domingo City 29 August, 1916. Lieutenant Jones did everything possible to get the engines and boilers ready, and if the elements that burst upon the vessel had delayed for a few minutes, the engines would have saved the vessel. With boilers and steam pipes bursting about him, in clouds of scalding steam, with thousands of tons of water coming down upon him and in almost complete darkness, Lieutenant Jones nobly remained at his post as long as the engines would turn over, exhibiting the most supreme unselfish heroism which inspired the officers and men who were with him. When the boilers exploded, Lieutenant Jones, accompanied by two of his

shipmates, rushed into the fire rooms and drove the men there out, dragging some, carrying others to the engine room where there was air to be breathed instead of steam. Lieutenant Jones' action on this occasion was above and beyond the call of duty."

Duty assignments followed at the Westinghouse Machinery and Electrical Company in East Pittsburgh and the Bureau of Engineering. In 1918 he was designated for the performance of Engineering duty. He later became assistant Naval Attache at London, England, with additional duty at Paris, Rome, Berlin, and The Hague. Upon his return he again did duty in the Bureau of Engineering until 1930, when he served on the Staff of the Commander, Battleships, Battle Force and later on the Staff of The Commander in Chief, Pacific Fleet. In 1931 he returned to the Bureau of Engineering as head of the Design Division, and in 1935 became Assistant to the Chief of Bureau. In 1936 he was appointed Naval Inspector of Machinery at the Electric Boat Company, Groton, Conn. He returned to Washington and the Bureau of Ships in 1940, where he was head of the Shipbuilding Division, with special responsibility in connection with the Shipbuilding program. In 1942 he was transferred to the Office of Procurement and Material as Assistant Chief. For service during this assignment he was awarded the Legion of Merit, with the following citation:

"For exceptionally meritorious conduct in the performance of outstanding services to the Government of the United States while associated with the Navy War Production Program since its inception in June, 1940, and as Assistant to Chief in Charge of Procurement and Material from October 17, 1942, to September 15, 1944. Challenged by many difficult organizational problems vitally affecting the speed and success of the Navy programs, Rear Admiral Jones has made his decisions with courage and followed them through with sound business judgment, launching and expediting production in the greatest Naval shipbuilding project in the Nation's history. Inspiring respect and confidence in his many contacts with industry and other Government Departments, he has served the interests of his country loyally and with untiring devotion to duty."

Concurrently, in 1942, the University of West Virginia awarded Admiral Jones the honorary degree of Doctor of Science.

In September, 1944, he was appointed Director of the Naval Engineering Experiment Station at Annapolis, where he served until his retirement in January, 1946. At that time he returned to Charleston, West Virginia, to make his home and enjoy his long time hobbies of fishing, hunting, and gardening, as well as a variety of other interests.

Admiral Jones was active for many years in the Society, as a member of the Council, as a contributor to the JOURNAL, in conjunction with other members, of articles on fuel oil and feed water, and as President of the Society in 1941. He was also a member and current Vice-President of the Society of Naval Architects and Marine Engineers.

Admiral Jones' decorations and medals included, in addition to those previously mentioned, the following:

> Cuban Pacification Medal, Mexican Service Medal, Victory Medals in both world wars, and the American Defense medal. He also held the expert team rifleman's gold medal, expert rifleman's medal with bars, and the National Team Rifleman's medal, all earned by his markmanship during the period 1907–1910.

Admiral Jones is survived by his widow, Mrs. Margaret Cox Jones, his son and daughter, Commander Frank C. Jones, U. S. N., and Mrs. M. J. Wyllie, and his sister, Miss Ida D. Jones. It is certain, too, that the many personal and professional friendships he formed in his lifetime can never die, and serve as no other monument ever can to mark the passing of an esteemed gentleman and engineer of truly great spirit and ability.

Thomas B. Stillman.

Thomas B. Stillman, aged 58, Engineering Consultant for The Babcock & Wilcox Company, and internationally known authority in the field of fuel-oil burning and marine boiler design, died suddenly on September 28 in Philadelphia while enroute to the Naval Boiler and Turbine Laboratory at the Philadelphia Navy Yard. His home was at 221 Coudert Place, South Orange, N. J.

Mr. Stillman was born in Hoboken, N. J., the son of Professor and Mrs. T. B. Stillman. His father was Professor of Engineering Chemistry at Stevens Institute of Technology. He received his engineering education at Stevens, being graduated with the degree of Mechanical Engineer in 1911. After graduation he entered the employ of The Babcock & Wilcox Company as an apprentice engineer, where he was ultimately assigned to the Marine Department. He became Application Engineer in 1931, and Engineering Consultant in 1948.

Mr. Stillman made many significant contributions to the development of modern boilers and fuel-oil burning equipment in naval and merchant vessels, holding more than 25 patents relating to boilers, superheaters, oil burners and economizers. In the field of oil burning equipment he was regarded as one of the world's outstanding authorities. He was the author of many technical papers in the field of oil burners and marine boilers.

Announcing Mr. Stillman's death, Alfred Iddles, President of The Babcock & Wilcox Company, said: "His work in the development of marine boilers and allied equipment contributed to the effectiveness of our navy and merchant marine in two world wars."

Mr. Stillman had been a member of The American Society of Naval Engineers since 1921. He was also a member of The American Society of Mechanical Engineers, American Society of Naval Architects and Marine Engineers, Theta Xi fraternity, the Railroad and Machinery Club, New York, and a Fellow of the American Association for the Advancement of Science. He was the Second Vice President and a member of the executive committee of the Alumni Association of Stevens Institute.

He is survived by his second wife, the former Eloise Brown Keenan. His first wife, the former Harriet T. McCormick, died in 1946. Also surviving are his sister Mrs. E. A. Quarles of Plainfield, N. J., and his brother, Albert L. Stillman, also of Plainfield.

BOOK NOTICES.

BATTLEFRONTS OF INDUSTRY, by David O. Woodbury. Published by John Wiley & Sons, Inc., New York, and Chapman & Hall, Limited, London, 1948. 333 pages. $3.50.

MEN AND VOLTS AT WAR, by John A. Miller. Published by McGraw-Hill Book Company, Inc., New York, London, 1947. 255 pages. $3.75.

These two books have come to the attention of the editor recently. They tell the World War II stories of two of the larger industrial organizations in the United States.

This notice is not offered to extol the two corporations concerned nor as an advertisement. There is no question that their contributions to democratic victory were remarkable. So were the contributions of the vast majority of the various and varied segments of United States Industry. Rather, this is written to call the attention of those interested in the mobilization of industry for support of any future effort of rallying to the defense of the United States to a very realistic phase of such mobilization.

These two books point up what actually happens in industry in a country like the United States when it is engaged in global war. It must become apparent to any thoughtful reader that any industrial mobilization plan must recognize that role of industry which is to wage a continual contest to force and use science and engineering and to convert their fruits into useable munitions, all against, not the calendar alone, but even against the clock.

ENGINEERING THERMODYNAMICS, published by International Textbook Company. Commander Alexander H. Zerban, U.S.N.R., a member of the Society is a co-author of this text book.

NEW MEMBERS

The Society has had the privilege of accepting the following new members since publication of the list in the August, 1948, issue of the Journal.

NAVAL.

Burrous, Robert G., Chf. Engineer,
Victory Lykes Bros. S. S. Co., New Orleans, La.
Mail Box 383, Greenville, N. C.

Comassar, S., Lieut. j.g., U.S.N.R.,
Instructor in Mechanical Engineering, Syracuse University,
Room 119 T. R., Syracuse U., Syracuse, N. Y.

Dahlhaus, F. J., Lieut., U.S.N.R.,
Designer Office Layouts, Shaw Walker Co.,
Mail 668 W. Sheridan Road, Chicago 13, Ill.

Downer, Hugh C., Lt. Comdr., U.S.N.R.,
Marine Supt., Picklands Mather & Co.,
2000 Union Commerce Bldg., Cleveland 14, Ohio.

Eikel, Robert, Commander, U.S.N.R.,
625 Cotton Exchange Building,
Houston 2, Texas.

Feinberg, Joseph L., Ensign, U.S.N.,
88 School St.,
Fall River, Mass.

Garren, H. L., Jr., Lieut. j.g., U.S.N.,
U.S.S. *Hyman* (DD 732) c/o Fleet P. O.,
New York, N. Y.

Jenkins, William H., Commander, U.S.N.R.,
c/o Pennsylvania Electric Co.,
Johnstown, Pa.

Kanakanui, William A., Lt. Commander, U.S.N., Ret.,
3035 Kiele Ave., Honolulu, T. H.

Kline, Edward C., Capt., U.S.N.R.,
The Electric Storage Battery Co.,
Allegheny Ave. & 19th St., Philadelphia 32, Pa.

Lewis, Paul Stewart, Lieut., U.S.N.R.,
Chem. Engr. Research & Development,
U. S. Bureau of Mines,
Mail 1232 Prescott St., McKeesport, Pa.

McGuigan, Joseph Lloyd, Captain, U.S.N.,
Bureau of Engineering, U. S. Maritime Commission,
Room 4863 Commerce Building, Washington, D. C.

Reeder, William C., Ensign, U.S.N.,
U.S.S. PC 154
Mail 811 Rexford Drive,
Beverly Hills, Calif.

Stekson, Alexander J., Lieut., j.g., U.S.N.R.,
Western Electric Co., Inc.,
Mail 9161 86th St., Woodhaven, L. I., N. Y.

Szczepkowski, Victor C., Lieut., j.g., U.S.N.R.,
106 Harding St.,
Bakersfield, Calif.

Thorpe, Harlan M., Captain, U.S.N.,
Staff Com. Bat. Crulant,
Building N-23, Naval Base, Norfolk, 11, Va.

Toutant, William T., Ensign, U.S.N.,
19 Logan St.,
Auburn, N. Y.

CIVIL.

Bennett, Arthur R., Marine Engineer, Bu. Ships, Navy Dept ,
Mail 2410 20th St., N.W.,
Washington, D. C.

Dashefsky, George J., Consultant in Mech. Engrg. &
Applied Mechanics,
Material Laboratory, Naval Shipyard, New York,
Brooklyn, N. Y.

Dudley, Sidney A., Mech. Engr.,
Bath Iron Works,
1027 Washington St., Bath, Maine.

Fischer, Frederick Kapp,
Mgr. Special Products Engineering,
Westinghouse Electric Corp., Lester, Pa.

Front, Richard W., Chief Engineer,
Henschel Corp., Amesbury, Mass.

Hayes, William A., Mgr. Govt. Section,
Westinghouse Electric Corp.,
Bloomfield, N. J.

Hunsinger, George O., National Electric Products Corp.,
338 14th St., Ambridge, Pa.

Jorgensen, Paul E., Marine Engr. P6,
Supt. Boiler Div., Boiler & Turbine Laboratory,
Naval Shipyard, Philadelphia, Pa.

Mourer, Peter W., Jr.,
P. O. Box 76,
Mercer Island, Wash.

Strope, Walmer Elton, Head Engineering,
Applications DN, U. S. Naval Radiological Laboratory,
San Francisco Naval Shipyard,
San Francisco 24, Calif.

Tucker, Warren R., Vice President,
Hydraulic Press Mfg. Co.,
218 S. Main St., Mt. Gilead, Ohio.

ASSOCIATE.

Graesser, Carl Herman, Vice Pres., Engrg.,
Manning, Maxwell & Moore, Inc., 11 Elias St.,
Bridgeport, Conn.

Horton, George, Surveyor Pacific Island Engrg. Co.,
Station 10,
Guam, Guam.

Hynson, Henry L., Sales Engineer,
Electric Storage Battery Co.,
1819 L St., N.W., Washington, D. C.

MR. ARTHUR G. FESSENDEN,

who, in October 1948, completed forty consecutive years as Clerk of the Society. The Secretary-Treasurers who have served in that capacity during these four decades know Mr. Fessenden's very real contribution to the success of every activity of the Society.

ASSOCIATION NOTES.

ORIGINAL ARTICLES.

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
Bureau of Ships, Navy Department
Washington 25, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is

received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Subject Matter and Authors.

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries

(*a*) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

Photographs.

Photographs of current or historical interest to readers of the Journal are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

Notice of Deaths.

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be greatly appreciated if any member who learns of the death of a member will advise the Secretary-Treasurer.

Prize Essay Contest 1949.

The Council has decided to continue the prize essay contest initiated in 1947.

Entries will be considered whether submitted by members or non-members of the Society.

No subject is specified. An essay on any subject which is of interest in the broad profession of Naval Engineering will be considered. Preference in judging winners will be given to essays of a thought provoking nature over technical articles which merely report an accomplishment or recommend an improvement of a limited nature. This matter must, of course, be left to the decision of the judges and should not discourage submission of any article.

All normal rules (see above) for submitting articles apply plus the requirement that the name of the author and title of the article appear on a detachable sheet of paper and that the author's name not appear on any manuscript page.

The mailing envelope must be plainly marked "ASNE Prize Essay Contest 1949".

All articles submitted must be received in Washington, D. C., prior to midnight (2400), 28 February, 1949.

Judging of the contest will be by three independent Naval engineers, none of whom will be an officer of the Society.

If, in the opinion of the judges *and* the Council, essays warrant, prizes of $500.00, $300.00 and $200.00 will be awarded. Any or all prizes may be omitted if the best articles submitted are not deemed worthy of awards of the values stated above.

Prize winning essays will be published in the JOURNAL without any additional compensation to the author. 50 reprint copies will be furnished the author free of charge.

Any non-prize-winning article submitted may be accepted for publication at the option of the Council. Regular rates will be paid in these cases.

Scholarship.

The Council has decided that the contract with Webb Institute of Naval Architecture shall be cancelled. The income of the

Society is not sufficient to permit continuing this arrangement whereby the Society contributes $2000.00 per year. Webb Institute is still available to any young man who can qualify for entrance, but after this year no student there will be identified as an A.S.N.E. scholar.

1949 Banquet.

Reservations have been made to hold the annual banquet of the Society on 29 April, 1949, at the Hotel Statler in Washington. Detailed announcement will be mailed to all members very soon.

Mr. Charles E. Wilson, Chairman of the Board of the General Electric Company has accepted an invitation to deliver the principal civilian address. Vice Admiral E. W. Mills, U.S.N., Chief of the Bureau of Ships, will act as Toastmaster. The principal Naval speaker will be announced at a later date.

Council Changes.

Incident to his detachment from duty in Washington, Captain Leroy V. Honsinger, U.S.N., tendered his resignation as a member of the Council, effective on his detachment about 5 November, 1948, Captain P. D. Gold, U.S.N. was named by the Council to complete Captain Honsinger's term which expires on 31 December, 1949.

Annual Meeting.

The annual meeting of the Society was held in the Navy Department on 5 October, 1948. The attendance was somewhat larger than usual in former years.

The report of the nominating committee was received and approved. Ballots have been mailed.

Other matters set forth in this section were reported and discussed.

The only new business was adoption of a motion to make reservation at this time for the 1950 banquet to assure that accommodations shall be available. A reservation for 21 April, 1950 has been made.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.